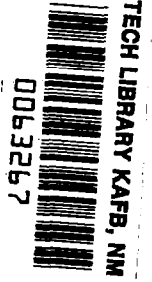


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Review and Status of Liquid-Cooling Technology for Gas Turbines

G. James Van Fossen, Jr., and Francis S. Stepka

APRIL 1979





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Review and Status of Liquid-Cooling Technology for Gas Turbines

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and Space Administration

**Scientific and Technical
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1979



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SUMMARY

A review was made of basic and applied research related to the liquid cooling of gas turbines along with an assessment of the state of the art and future technology requirements. Although a number of liquid-cooled turbine demonstrators have been designed, built, and tested over the years, the designs were generally based on little information and gross approximations. The results of these demonstrations generally disclosed problems, questions, and awareness that additional design information was needed.

The review has indicated a lack of adequate basic data to provide understanding of the heat transfer, pressure drop, critical heat flux, and flow stability in a cooling passage of a rotating turbine blade. It has also indicated that existing technology is not adequate to design a liquid-cooled turbine with a high assurance of success or to optimize liquid-cooling systems for a given turbine application. Considerable basic and applied research on liquid cooling of gas turbines is still required. The report addresses specific areas where research is required.

INTRODUCTION

The major advancements and applications in cooled gas turbine engines to date have been with direct air cooling of the hot section parts. This has been due to the dominant influence of aircraft gas turbines wherein air cooling has been more attractive. Mechanical complexities associated with liquid-cooling systems along with inadequate heat-transfer and coolant-flow data have contributed to the lag in its application to gas turbines.

The recent concern with our limited petroleum supply and the desire to reduce fuel consumption have rekindled interest in liquid cooling and its potential for permitting higher maximum gas temperatures, greater cycle efficiencies, and higher specific work outputs. Improvements in fabrication technology and nondestructive inspection techniques have added to the interest. The analyses described in references 1 and 2, for example, show the potential benefits of using water-cooled turbines in electric utility applications. The combined gas turbine — steam cycles studied in the references used water-cooled gas turbines and had higher thermal efficiencies and specific power outputs than those with current or advanced air-cooled gas turbines. The resultant potential benefits are conservation of fuel usage, greater flexibility in the fuel used, and reductions in the cost of electricity.

As a consequence of this renewed interest, a review of basic and applied research related to the

liquid cooling of gas turbines has been conducted. The methods and information reviewed are more applicable to ground-based turbines such as those intended for electric utility peaking or combined cycle use. This is because the engine weight and design considerations used to incorporate liquid cooling into ground-based applications are less critical than those used in aircraft applications. The information contained in this report, however, is generally applicable to other gas turbines. The report reviews and discusses liquid-cooling methods and systems for turbines and the results from demonstrator turbines that have been built and tested. The physical mechanisms underlying heat transfer, pressure drop, flow stability, and critical heat flux of selected systems are also discussed. An assessment has been made of the state of the art and future requirements. Although personal inquiries on research and technology were made as late as mid-1977, the survey herein includes only available information published prior to that time.

A comprehensive bibliography of research related to the liquid cooling of gas turbines is also presented.

REVIEW OF PAST EFFORTS

Cooling Methods and Demonstrator Turbines

The liquid-cooling schemes that have been tried or proposed are reviewed in this section. Rigs or turbines that have been built and tested to evaluate the feasibility of these schemes are discussed concurrently. Rotor blade cooling is emphasized because the cooling of stationary parts is considered to be nearly within the state of the art; for completeness, liquid-cooled stator tests are also discussed. The physical mechanisms underlying heat transfer, pressure drop, and flow stability of selected systems are discussed in more detail in the Basic and Supportive Research section.

Forced Convection

The first method to be discussed, simple forced convection, is shown schematically in figure 1. Water taken on board the rotating turbine wheel through a seal moves radially outward along the disk and then circulates through cooling passages in the blade. The coolant then moves radially inward along the disk and exits through another seal. In this case forced convection is actually a misnomer; because of the density difference between the cold inlet tube and the hot outlet tube, the flow is driven by natural convection. Theoretically,

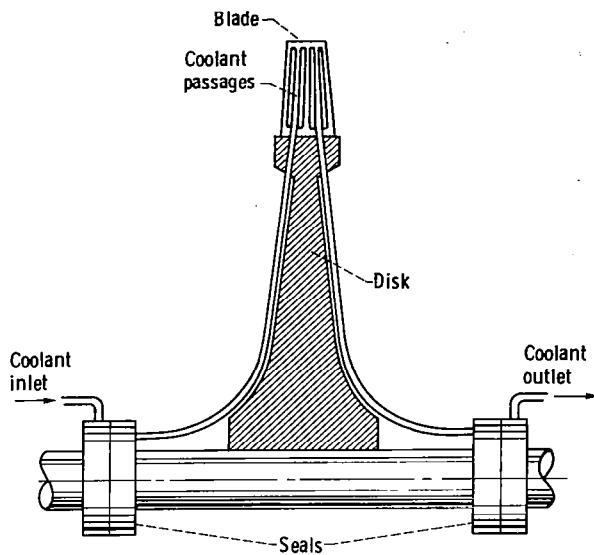


Figure 1. - Simple forced convection.

almost any coolant can be used with this system; however, water or steam is probably the only practical choice because of the possible leakage and quantities required. The main advantage of the forced-convection liquid-cooling method is the potential for obtaining low metal temperatures. This could be important to turbine hot section life if the heavier ash bearing fuels are used. The ash formed by the combustion of some fuels will form a coating on the hot parts. The ash contains sodium compounds which remove the oxide layer from protective coatings; the sodium compounds also depress the melting temperature of the superalloy material. Thus, if the hot part is not kept below this depressed melting temperature, corrosion of the molten surface layer will proceed rapidly. There is no agreement as to what the hot part surface temperature should be, but it is thought to be in the 756 to 1089 K (900° to 1500° F) range. Some other advantages of this system are the simplicity and easy recovery of heat from spent coolant. The main disadvantage of this system is the high coolant pressure within the blade if dense liquids such as water are used. For a typical, large, ground-based, electric utility gas turbine having a diameter of 2.07 meters (6.8 ft) and turning at 3600 rpm, the pressure at the blade tip would be over 75.845×10^6 pascals (11 000 psi). Another disadvantage of this scheme is that a high water flow rate is required to keep the coolant from becoming saturated and prevent net boiling. Also, foreign matter in the coolant could be centrifuged to the blade tips and block the cooling passages.

Several liquid-cooling demonstrators have used forced-convection systems. Figure 2 shows an

aluminum forced-convection water-cooled turbine blade tested at the NACA Lewis Flight Propulsion Laboratory. This work is described in Kottas and Sheflin (ref. 3), Bartoo and Freche (ref. 4), Freche and Diaguila (ref. 5), and Ellerbrock (ref. 6). In operation, water was pumped out the two radial passages nearest the leading edge. Near the blade tip drilled crossover passages conducted the water to the two radial trailing edge passages. The spent cooling water was taken off the rotor through the axial discharge hole shown in figure 2. Because the coolant outlet is at a larger radius than the inlet, work was taken off the wheel to pump coolant. Blade temperatures were kept extremely low; the highest measured temperature was 470 K (386° F). The maximum turbine inlet gas temperature was 1422 K (2100° F). The small number of relatively large cooling passages lead to high thermal gradients. Clogging of cooling passages was not found to be a problem, and no structural failures developed. The tests, however, were of short duration.

Figures 3(a) and (b) show the forced-convection cooling scheme used in an engine built by the Solar Aircraft Company. This work is described in references 7 to 9. The single-stage water-cooled turbine is shown in figure 3(a), and the water-cooled blade is shown in figure 3(b). The blade was made from a

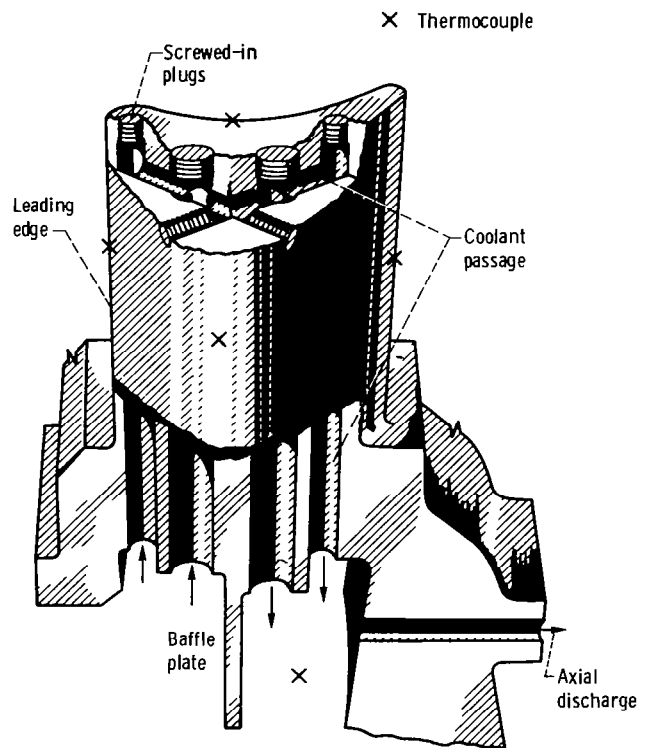
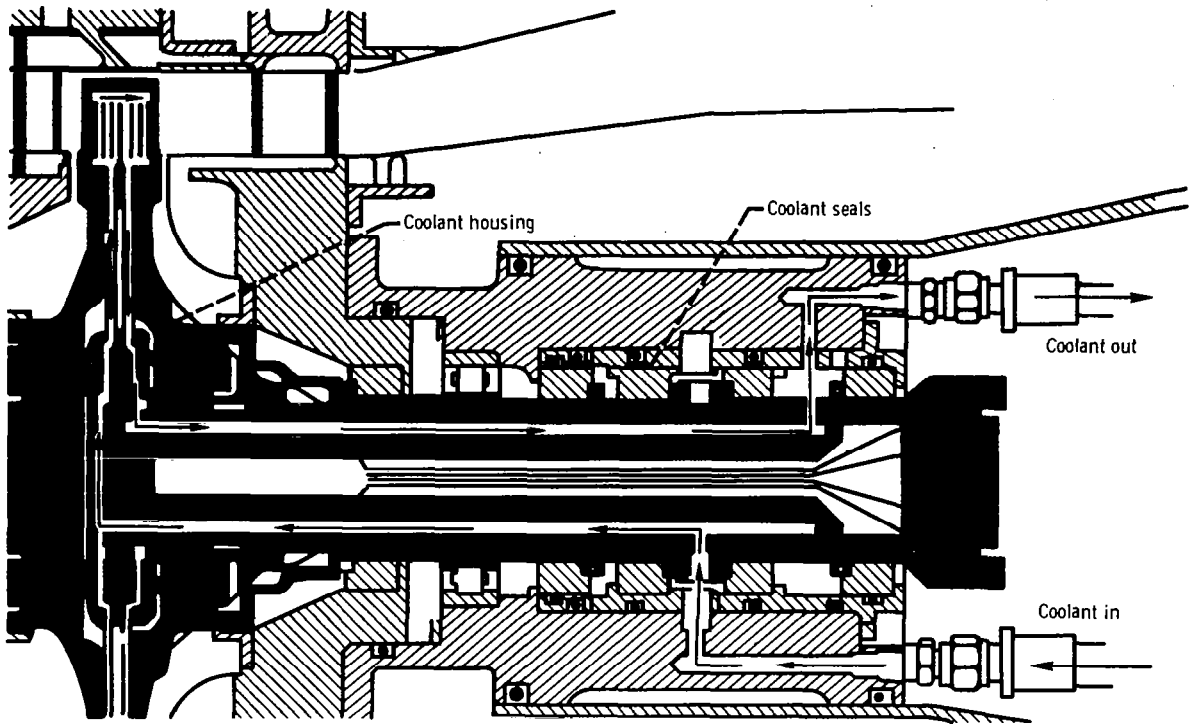
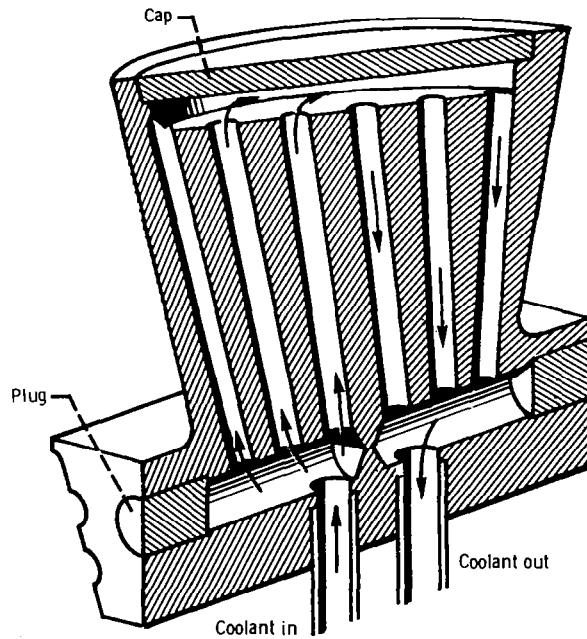


Figure 2. - NACA forced-convection water-cooled aluminum turbine (ref. 3).



(a) Solar forced-convection single-stage water-cooled turbine.



(b) Blade cooling passages for Solar water-cooled turbine.

Figure 3. - Solar Aircraft Company forced-convection-cooled turbine (ref. 7).

low alloy steel with drilled radial cooling passages that were sealed at the tip by a welded tip cap. In highly stressed regions, the blade temperatures were kept below 700 K (800° F) with peak temperatures less than 867 K (1100° F) and the cooling water temperature was kept below 367 K (200° F). The turbine inlet temperature was 1228 K (1750° F) at a 4.87 to 1 overall pressure ratio. The large amount of heat removed from the cycle by the cooling water lowered the efficiency by an estimated 20 percent. The water-cooling scheme was successfully demonstrated, but blade tip cap failures indicated improved fabrication and inspection techniques were required.

Two variations of the simple forced-convection cooling configuration are shown in figures 4(a) and (b). The first is forced convection with trailing edge ejection. Water is taken on board the wheel through a seal as before, but after circulating through the blade cooling passages it is discharged through the gas stream through orifices in the blade trailing edge. Although this configuration avoids the outlet seal, there are disadvantages. If the passages are filled with water, the high pressure at the tip still exists and the resulting high pressure drop across the discharge orifices will lead to high velocities and erosion of these orifices. Heat added to the coolant is not recoverable; in fact, the spent water lowers the gas stream temperature entering the next turbine stage. Any water not flashed to steam will impinge on downstream stages and cause erosion. No demonstrators have been built with this cooling scheme. The second variation, shown in figure 4(b), is forced convection with tip ejection and collection. Coolant is taken on board the wheel through a seal or other means and, after cooling the blade, is ejected at the tip. Some of the water in the blade can be allowed to vaporize, thus taking advantage of the heat of vaporization to lower coolant-flow requirements. At the stationary shroud a collector catches the unvaporized water that is slung from the blade tip. The ejected steam enters the gas stream. Advantages of this system are low coolant passage pressure, lower potential for cooling passage clogging because of tip ejection, and lower water consumption. Other advantages are potential recovery of waste heat from the collected coolant and elimination of one of the rotary seals. Disadvantages include erosion of the stationary shroud from water droplets, work taken off the wheel to pump coolant from a low to a high radius, erosion of downstream stages from uncollected water drops, and lowering the gas temperature entering downstream stages due to the mixing of steam and uncollected water. The collection of high velocity water droplets is not easy; the potential for erosion of the stationary shroud, collection slot, and downstream turbine stages is great. A disadvantage of both of these systems is that a relatively large quantity of high quality coolant is thrown away; this disadvantage would probably preclude using this method in aircraft applications.

Figure 5 shows the cooling concept used by General Electric and described in references 2, 10, and 11. This system is designed to be used in a combined gas turbine — steam cycle for generating electricity from coal-derived fuels. Spraying cooling water from stationary nozzles into the rotating gutter (shown in fig. 5) eliminates the need for a rotary seal. From the rotating gutter the cooling

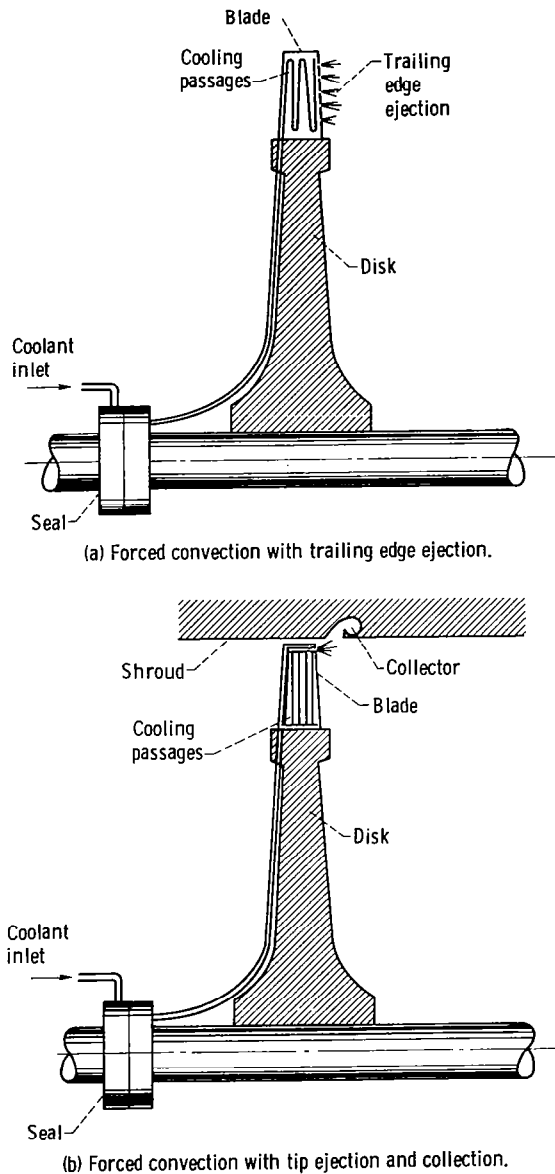


Figure 4. - Variations of forced-convection cooling scheme.

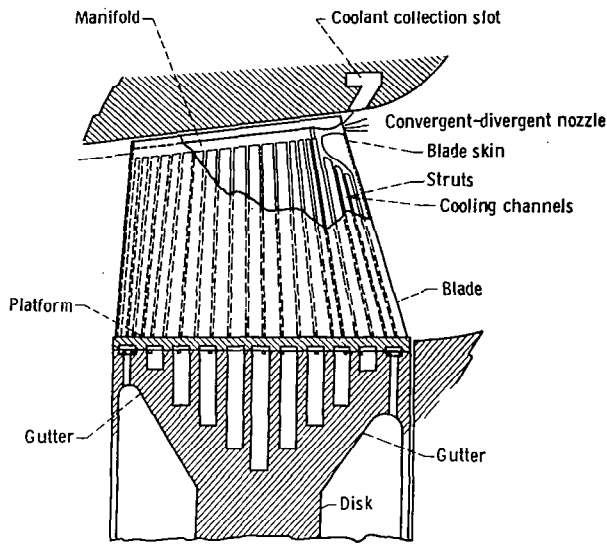


Figure 5. - General Electric ultra-high-temperature-turbine cooling scheme.

water is routed into the cooling passages. The cooling water flow is controlled by a system of weirs at the base of the blade; this system is described in detail in Kydd (ref. 12), Moore (ref. 13), and Gron-dahl and Eskesen (ref. 14). From the blade cooling passages water then enters a manifold and exits at the blade tip through the converging-diverging nozzle. The flow rate is low enough that some of the water evaporates and enters the gas stream as steam; some of the remaining water is supposed to be caught and collected by the slot in the casing. This cooling concept minus the collection device has been successfully tested to as high as 1830 K (2850° F) at 16 atmospheres in a 24.7-centimeter-(9.7-in.-) diameter demonstrator turbine.

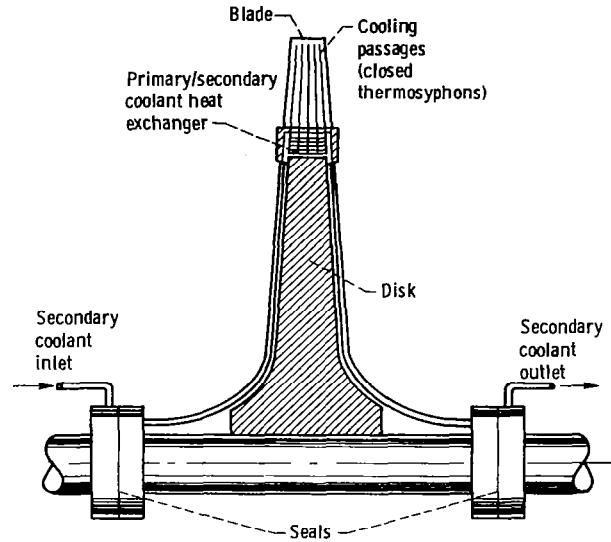
In summary, forced-convection water cooling can be a relatively simple and effective means for cooling high temperature gas turbines. Tip bleed of the water appears to be an effective method if an abundant supply of coolant exists such as in a ground-based machine. The tip bleed configuration avoids high internal fluid pressures and may eliminate some internal clogging problems. The method presents problems of recovery of the water and erosion by the exiting liquid. A major advantage of the forced-convection systems is the potential to cool hot parts to very low temperatures, which may provide protection from erosion and corrosion caused when heavier ash bearing fuels are used.

Thermosyphon

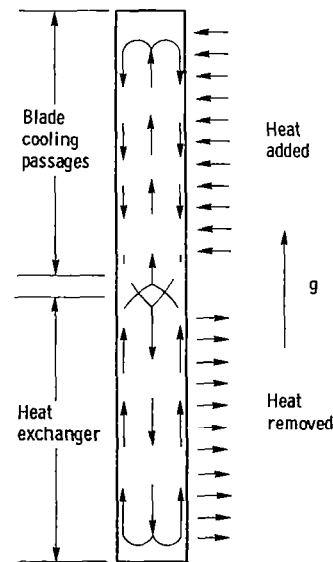
In a thermosyphon heat is transported by moving a fluid, the fluid motion being caused by a density difference in a body force field. In the case of the turbine wheel, the body force field is the centrifuga-

force field caused by rotation, and it is proportional to the product of the radius and the square of rotational speed. This can be as high as 20 000 g's for a typical turbine; thus, there is a large potential for heat transfer. There are three basic types of thermosyphons: the closed thermosyphon, the open thermosyphon, and the closed-loop thermosyphon.

Closed thermosyphon. — The closed thermosyphon is illustrated in figure 6(a) in a turbine application. The thermosyphons form the cooling passages in the blade: each cooling passage is a



(a) Possible gas turbine application.



(b) Closed thermosyphon tube.

Figure 6. - Closed thermosyphon system.

closed tube with the primary heat-transfer media (liquid or gas or both) sealed inside. Primary coolant near the tube wall is heated in the blade cooling passage; this hot, less dense, coolant flows inward toward the cool end and is displaced by cold, more dense fluid from the heat exchanger. This process is illustrated in figure 6(b). In the heat exchanger end of the tube, heat is removed from the primary fluid near the tube wall. This cool, more dense fluid moves outward toward the heated end. In the region between the heat exchanger and the blade cooling passage hot and cold fluids collide. Hot fluid flowing in annular-type flow in the cooling passage must find its way to the central core in the heat exchanger end and vice versa for the cold fluid in the heat exchanger. Several possible mechanisms for this transfer exist, and they are discussed in the Basic and Supportive Research section. A secondary coolant removes heat from the heat exchanger and carries it away. The secondary coolant could be water that is externally supplied, it could be the fuel, or it could be compressor discharge air that would be heated before entering the combustor. The latter two configurations would be applicable to aircraft systems, while water could be used for ground-based machines.

Another type of closed thermosyphon is the two-phase thermosyphon or vapor chamber. This type has a similar construction, but the chamber is only partially filled with the working fluid. At the design condition the working fluid is vaporized in the blade cooling passage and recondenses at the cooled heat-exchanger end. Condensate runs outward in a film from the condenser end to the heated end, thus cooling the blade.

Since the closed thermosyphon is completely sealed, clogging by foreign matter is not a problem; also, the coolant pressure within the blades can be maintained lower than that for some other systems. The choice of primary working fluid is not limited as with other systems; thus, liquid metals or boiling and condensing liquids with their excellent heat-transfer characteristics can be considered. On the negative side, should a crack in the blade material develop, primary coolant can be lost and cooling will no longer take place. The closed thermosyphon system requires a primary to secondary coolant heat exchanger. It may be difficult to find adequate space on the turbine wheel for this and it may also be costly. If the working fluid and the thermosyphon wall material are not compatible, corrosion could cause clogging or impede efficiency for long-term operation.

Cohen and Bayley (ref. 15) performed experiments on closed thermosyphon tubes in a rotating rig to simulate a turbine environment. The tubes had their axes oriented radially with respect to the

axis of rotation and were filled with various quantities of mercury. The outer half of each tube was heated with hot air, while the inner half was cooled with cold air. It was found that the heat-transfer rate was independent of the amount of mercury in the tube after a certain critical volume was reached. Bayley and Bell (ref. 16) used the same rig to test tubes filled with various quantities of mercury up to 440 g's. Their results were similar to those of Cohen and Bayley (ref. 15).

In the early 1960's the General Electric Company demonstrated a liquid-metal-cooled turbine rotor as reported by Suciú (ref. 17). Not much information is available on this work. All that is known

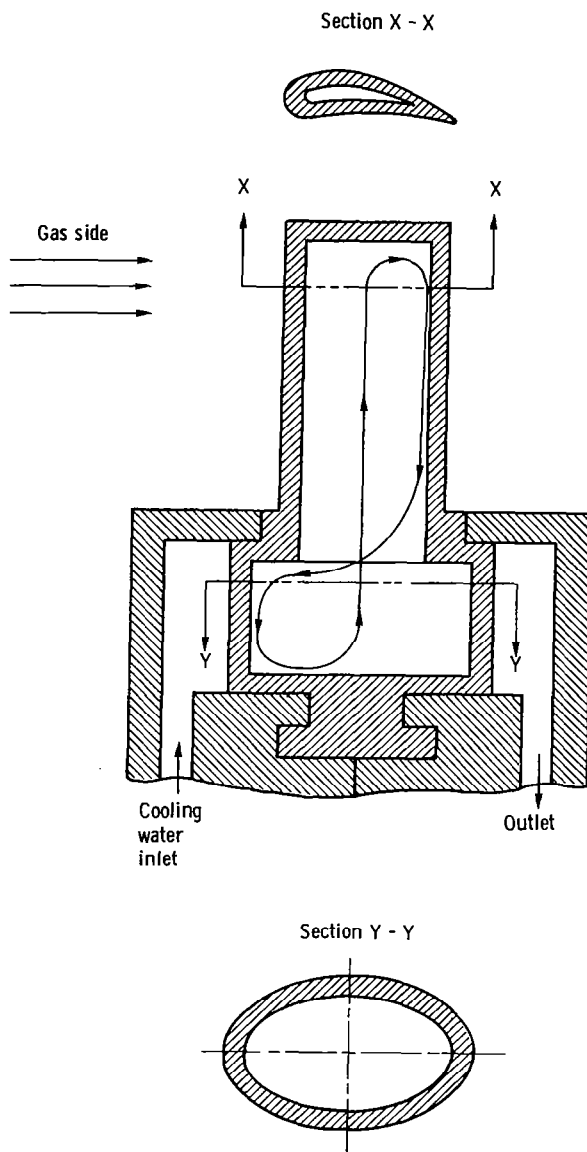


Figure 7. - Schematic diagram of semiclosed thermosyphon (ref. 20).

is that the blade was run at temperatures over 1478 K (2200° F) and was cooled by closed thermosyphons partially filled with a liquid metal. Heat was extracted at the blade base by an air heat exchanger. It was reported that the heat flux at the leading edge was increased by a factor of 4 over air cooling.

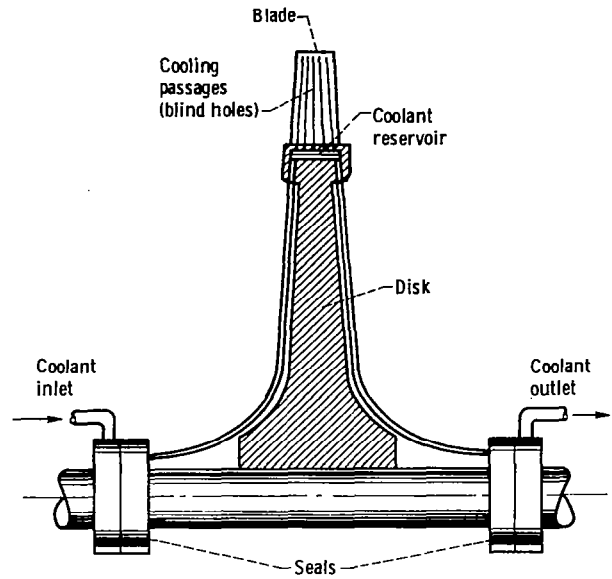
Genot and LeGrives (ref. 18) and LeGrives and Genot (ref. 19) found similar results in a rotating rig with a liquid-metal closed thermosyphon. They concluded that the closed thermosyphon could compete with film cooling as a viable cooling scheme for aircraft engines.

Ogale (ref. 20) ran experiments on a modified closed thermosyphon that he called a "semi-closed" thermosyphon. This device, illustrated in figure 7, had a larger hydraulic diameter at the cooled end than at the heated end; this was supposed to allow a shorter length heat exchanger section. The thermosyphon was filled with liquid metal and the secondary coolant was distilled water. The cross section of the heated end had a shape similar to that of a turbine blade. The experiments were conducted on a rotating rig to simulate the large centrifugal force field of a turbine. Overall heat-transfer coefficients were 3 to 4 times lower for the semi-closed thermosyphon than they were for the ordinary closed thermosyphon of other investigators.

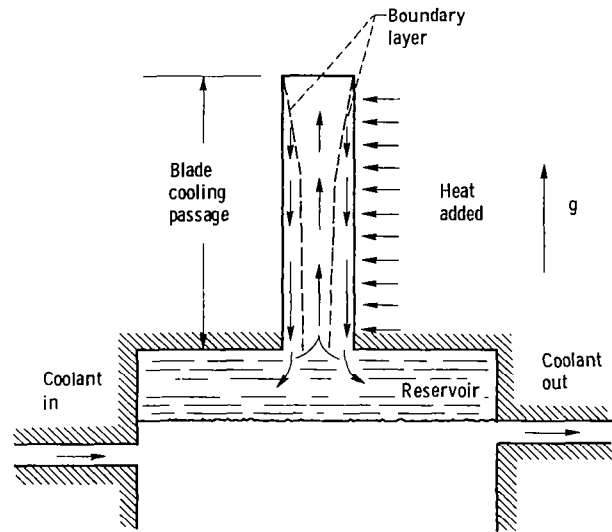
Open thermosyphon. — Figure 8(a) shows the open thermosyphon system, and figure 8(b) shows the cross section of a single cooling passage. The coolant enters a cavity or reservoir at the base of the blade. Cooling passages in the blade are open to this reservoir, and the fresh, more dense coolant in the reservoir displaces the hot fluid in the cooling passages. The hot fluid from the blade then mixes with the fluid in the reservoir that is continually being drained off and replaced with a cool fluid from the inlet.

Advantages of this system are its simplicity and that no heat exchanger is required. Fluid pressure at the blade tip is a function of the location of the free surface in the reservoir; thus, pressure within the blade can be lower than that for other systems. A disadvantage of the open thermosyphon is that foreign matter in the coolant can collect at the blade tips because the cooling passages are blind holes. If the length to diameter ratio of the thermosyphon is too large, performance will suffer because of the growth and mixing of the boundary layers of hot and cold fluids flowing in opposite directions. Bayley and Martin (ref. 21) state that water is the only practical coolant for the open thermosyphon system. Water, however, can be chemically active so corrosion could be a problem.

An open thermosyphon cooled turbine was



(a) Possible turbine cooling configuration.



(b) Single cooling passage.

Figure 8. - Open thermosyphon system.

designed by Ernst H. W. Schmidt during World War II in Germany. Test results on this turbine are reported in Smith and Pearson (ref. 22). The turbine operated for about 50 hours at 1367 K (2000° F) and achieved 2 hours of running at 1533 K (2300° F). The inlet pressure was only 1.2 atmospheres, which meant a relatively low heat load compared to a modern turbine. A four-stage turbine fabricated in Germany was taken to England and tested after the war. Results of the test are given in Robinson (ref. 23), and the operation is described in Schmidt (ref. 24). Figure 9 shows a cross section of the rotor.

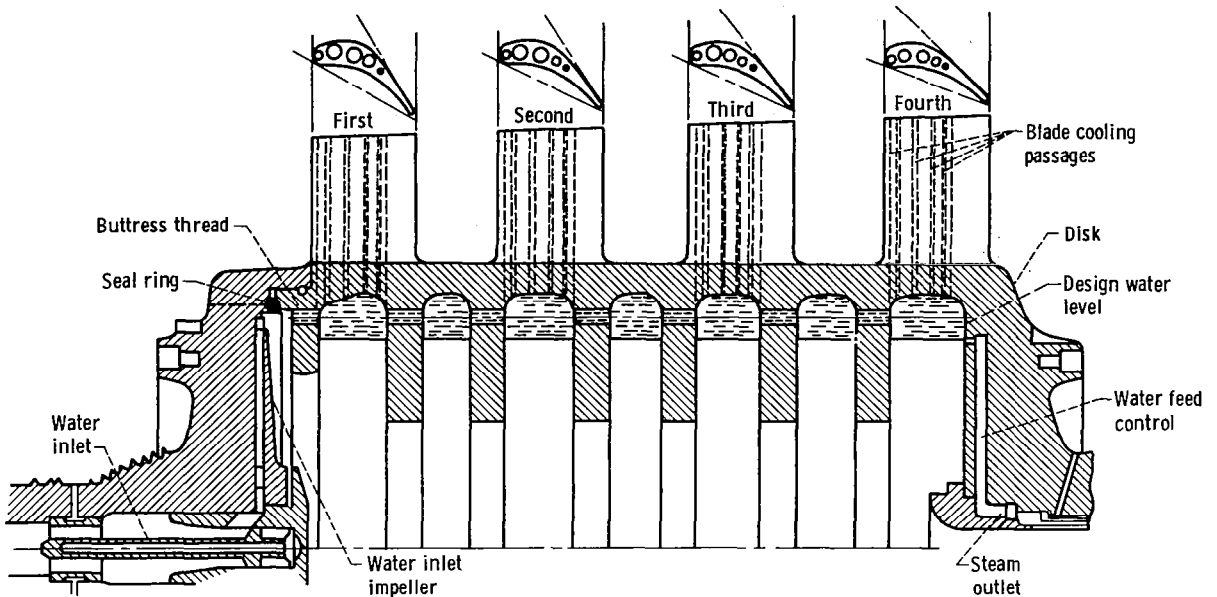


Figure 9. - Schmidt turbine (ref. 24).

The blade cooling passages are open thermosyphons, and the reservoir consists of a ring of water held against the hollow disk by centrifugal force. Hot water moves to the free surface of the water ring where it evaporates and carries away the heat. Steam thus generated is removed through a port at the drum centerline. Several problems developed during testing. The drum and blades were machined

from low alloy steel and were subjected to corrosion from the water coolant. When testing was completed, several cooling passages had been completely corroded through and most of the cooling passages were blocked with iron oxide. The turbine was tested to 923 K (1202° F), but it was only able to achieve 61 percent of design speed because of vibration. The source of vibration was not known, but it was speculated to be caused by boiling in the cooling passages or by instabilities in the water ring.

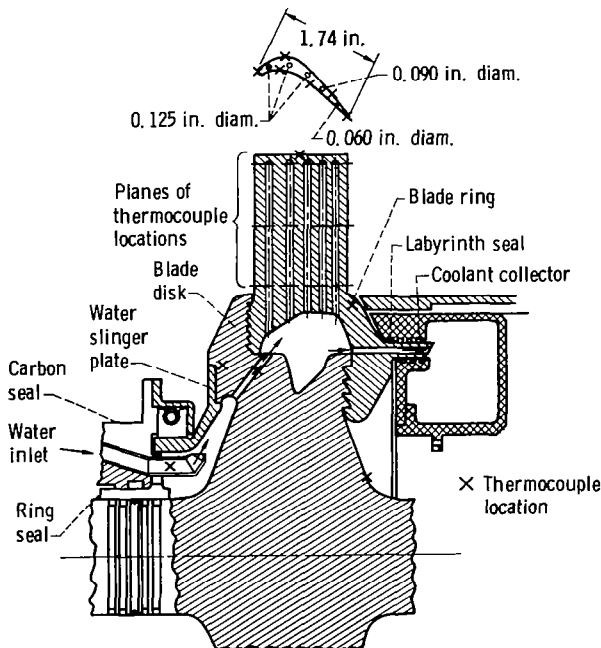
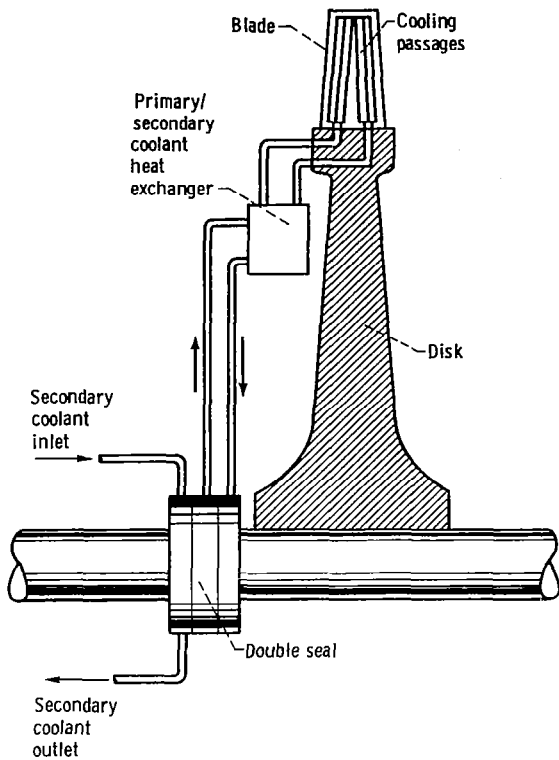
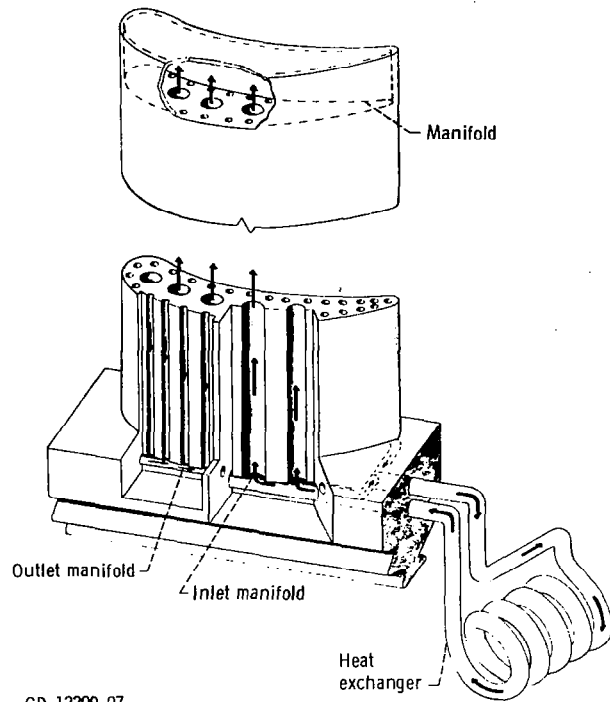


Figure 10. - NACA open-thermosyphon water-cooled turbine (ref. 4).

Figure 10 shows an open thermosyphon, water-cooled turbine blade tested at the NACA Lewis Flight Propulsion Laboratory. Results of the work are reported in Bartoo and Freche (ref. 4), Freche and Diaguilia (ref. 5), Ellerbrock (ref. 6), Freche (ref. 25), Freche and Schum (ref. 26), and Curren and Zalabak (ref. 27). As shown in figure 10, water was sprayed into a rotating gutter from stationary nozzles before it flowed through holes in the disk to a cavity or reservoir between the blade base and the disk. This water then circulated through open thermosyphons in the blades. The heated water from the thermosyphon then mixed with the cooler water in the reservoir at the blade base; this mixed coolant was continuously taken off the wheel through exit holes. The maximum gas temperature of 1222 K (1740° F) resulted in an average blade temperature of 450 K (350° F) and a maximum measured blade temperature of 739 K (870° F) at the trailing edge. It was found that increasing the coolant flow could lower the average blade temperature but not the maximum temperature. An improvement in the cooling effectiveness of passages



(a) Possible turbine cooling configuration.



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(b) Details of blade cooling passages and heat exchanger.

Figure 11. - Closed-loop thermosyphon system.

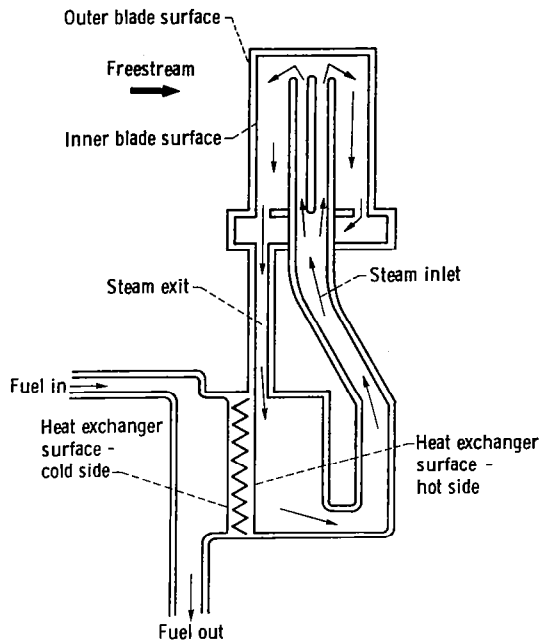
with large length to diameter ratios was obtained when they were connected by tip crossover passages to cooling passages with smaller length to diameter ratios as reported in Zalabak and Curren (ref. 28). This arrangement let cool fluid from the larger diameter passage flow unidirectionally down the small passage; the problem of hot and cold fluid flowing in opposite directions and mixing in long thin passages was relieved in this manner. This arrangement might be called an open-loop thermosyphon. The open-loop thermosyphon seems to be the most practical of the open thermosyphon configurations; however, no other known research has been conducted.

Closed-loop thermosyphon. — The third type of thermosyphon is the closed-loop thermosyphon shown in figure 11(a) in a possible turbine blade cooling configuration. Details of the blade cooling passages and heat exchanger are depicted in figure 11(b). Instead of the coolant flowing in two directions simultaneously in the cooling passages as in the closed or open thermosyphons, it flows in one direction in each passage. Cool, more dense fluid moves radially outward through the large central feed passages and is then manifolded at the tip to flow radially inward through the small cooling passages around the perimeter of the blade. Heated

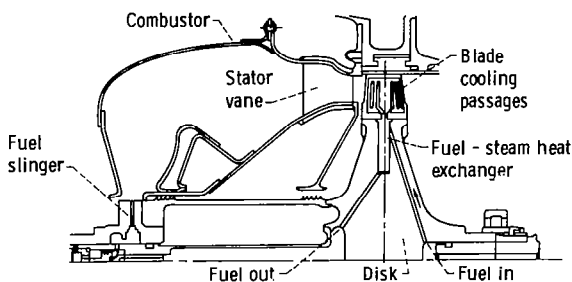
fluid in the cooling passages then moves inward to a heat exchanger where it is cooled by a secondary coolant.

The closed-loop thermosyphon has the same advantages and disadvantages as the closed thermosyphon. However, because flow is unidirectional in all the passages, much longer passages can be utilized. Long, thin passages may be necessary in the turbine application to minimize thermal gradients.

A closed-loop thermosyphon cooling scheme was demonstrated in an engine designed to run at 1533 K (2300° F) by the Continental Aviation and Engineering Corporation. The work is described in Gabel (ref. 29), Gabel and Tabbey (ref. 30), and Johnson (ref. 31). Figure 12(a) shows a schematic of the closed-loop thermosyphon. The primary coolant used was steam and the secondary coolant was fuel. As shown in figure 12(b), a cross section of the combustor and turbine, fuel is taken on board the wheel through a seal at the right end. The fuel then circulates through the heat exchanger that extends down past the blade base into a cavity in the disk. After removing heat from the steam, the fuel flows through the hollow shaft to the combustor where it is sprayed from a slinger and burned. The heat removed from the blades can thus be added back to



(a) Flow schematic for Continental turbine.

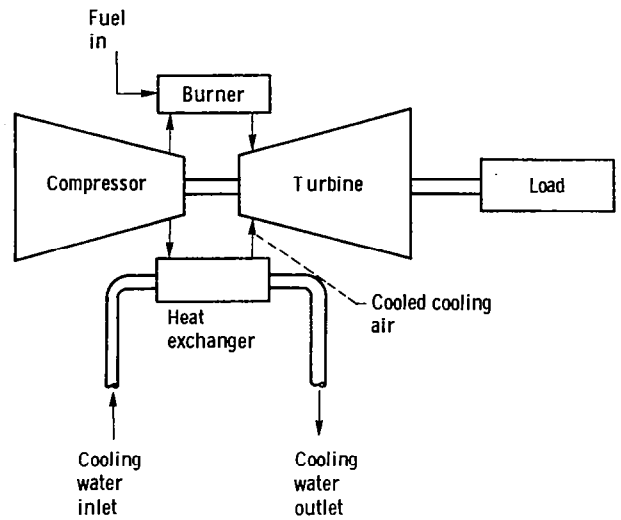


(b) Cross section of turbine and combustor.

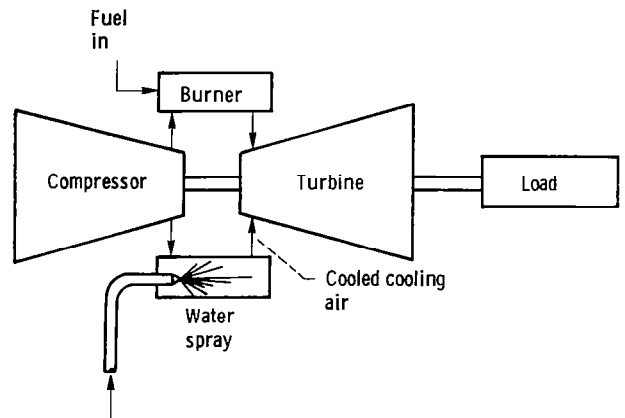
Figure 12. - Continental Aviation and Engineering Corporation closed-loop thermosyphon turbine demonstrator (ref. 30).

the cycle through the fuel. Several hundred hours of elevated temperature testing were accomplished with a maximum temperature of 1617 K (2450° F). Twenty-one hours were accumulated above 1422 K (2100° F). When the gas temperature was 1417 K (2090° F), blade temperatures were measured to be between 850 and 1072 K (1070° and 1470° F). Several blade failures were encountered: the cause of the blade failures was believed to be the loss of primary coolant through cracks in the blades or welds. Better nondestructive testing procedures might have eliminated this problem.

Summary. — Thermosyphons provide another means of cooling high temperature gas turbines. The closed-loop thermosyphon appears particularly attractive in systems where coolant cannot be thrown away as in aircraft applications. Another advantage of the closed-loop system is that it is free from clogging by foreign matter in the coolant. The



(a) Air-water heat exchanger system.



(b) Water spray system.

Figure 13. - Cooled cooling air configurations.

main disadvantage of all the closed systems is that if the primary coolant escapes through cracks all cooling stops. The open-loop and closed-loop thermosyphons are attractive for systems where long, thin passages must be used. The open-loop thermosyphon system is particularly attractive because coolant leaks in the blade would not lead to blade failure from loss of coolant. However, it does have the disadvantage of possibly being clogged by foreign material from the coolant.

Other Cooling Methods

Cooled cooling air. — A different method of using liquid to allow operation at higher turbine inlet temperatures is to lower the temperature of the compressor discharge air used for turbine cooling. Figure 13 shows two proposed methods of accomplishing this. In figure 13(a) compressor discharge air used for cooling the hot section components is

passed through a heat exchanger where it gives up heat to a secondary liquid coolant. The cooled cooling air then goes to conventional air-cooled vanes and blades. Advantages of this method are that state-of-the-art hardware can be used and that high liquid-coolant pressures and the potential for erosion by liquid are avoided. It may be feasible to use fuel to cool the cooling air; thus, the heat removed from the cooling air could be added back to the gas turbine cycle directly. This scheme could be readily adapted to aircraft application. Edwards (ref. 32) calculates that, if fuel were available at 423 K (301° F) at a fuel to air ratio of 0.025, a cooling air flow rate of 6 percent of the main flow could be cooled by 260 K (468° F), assuming the fuel could be heated to 646 K (703° F) without coking. It was assumed that for aircraft applications the fuel would be used as a heat sink for various engine accessories and would, therefore, already be heated to 423 K (301° F). If other liquids are used, it is possible that heat removed from the cooling air by the liquid could be used elsewhere in the cycle. The main disadvantage is the added complexity and cost of the heat exchanger.

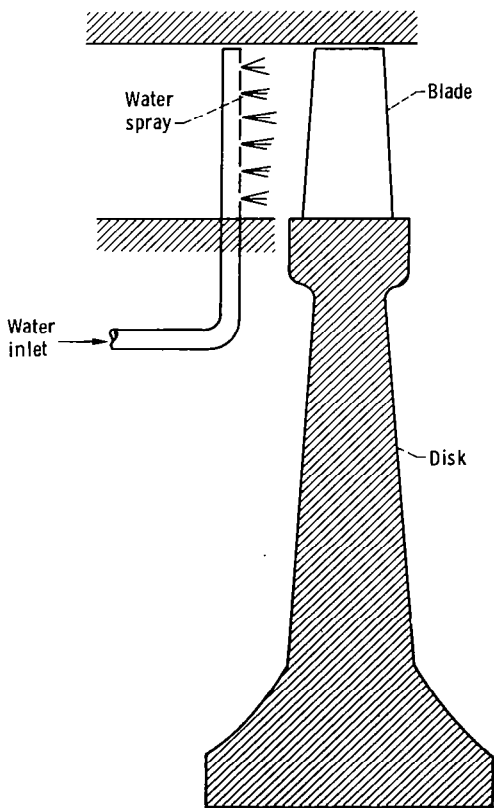


Figure 14 - Spray cooling configuration.

Edwards (ref. 32) also described the system shown in figure 13(b). In this system water is sprayed directly into the hot cooling air; this water vaporizes and cools the cooling air. Although this scheme would not be as practical for aircraft use, except for emergency or intermittent use, it would be simple to implement in a ground-based turbine where the extra weight of the water coolant would not pose a penalty. Edwards states that by injecting 10 percent water into the cooling air the temperature could be reduced from 700 to 440 K (800° to 332° F). This could allow the turbine inlet temperature to be increased by about 100 K (180° F). The obvious advantage of this system is the elimination of a costly heat exchanger. A disadvantage is that minerals in the water could clog the small cooling passages or film cooling holes; thus, demineralized or treated water should be used.

Goodyer and Waterston (ref. 33) proposed adding water mist to the cooling air to increase its cooling capacity as well as lower its temperature. They measured heat-transfer coefficients of an impinging air-water mist. The addition of 6 percent water by weight doubled the heat-transfer coefficient of an impinging air jet.

Spray cooling. — Spray cooling is illustrated in figure 14. Water is sprayed from stationary nozzles or nozzles in the rotating blade bases onto the exterior of the rotating blades. The evaporating water cools the blade surface. The spray cooling scheme was investigated at the NACA Lewis Flight Propulsion Laboratory and elsewhere to determine the feasibility of using it to allow increased turbine speed and temperature and thus increased thrust during takeoff. The work is reported in Freche and McKinnon (ref. 34), Freche and McKinnon (ref. 35), Burke and Kemeny (ref. 36), Freche and Hickel (ref. 37), and McKinnon and Freche (ref. 38). With a 1127 K (1568° F) turbine inlet temperature, blade temperatures of 418 K (292° F) were measured. The cyclic application of the water spray cooling technique caused thermal shock failure of the blades. Because of this, it was never used on an aircraft. The large quantities of water required make the system impractical for steady-state use.

Sweat cooling. — Sweat cooling, as the name implies, uses a liquid coolant that seeps through pores in the blade wall and evaporates from the blade surface. The air-cooled counterpart of sweat cooling is transpiration cooling. In order to control coolant flow distribution and flow rates, very fine pores would be required in the blade. These would be subjected to clogging from foreign matter in the coolant as well as plugging from corrosion. Sweat cooling has not been tried in a turbine application.

Summary. — Of the three methods discussed herein, the most practical is the cooled cooling air.

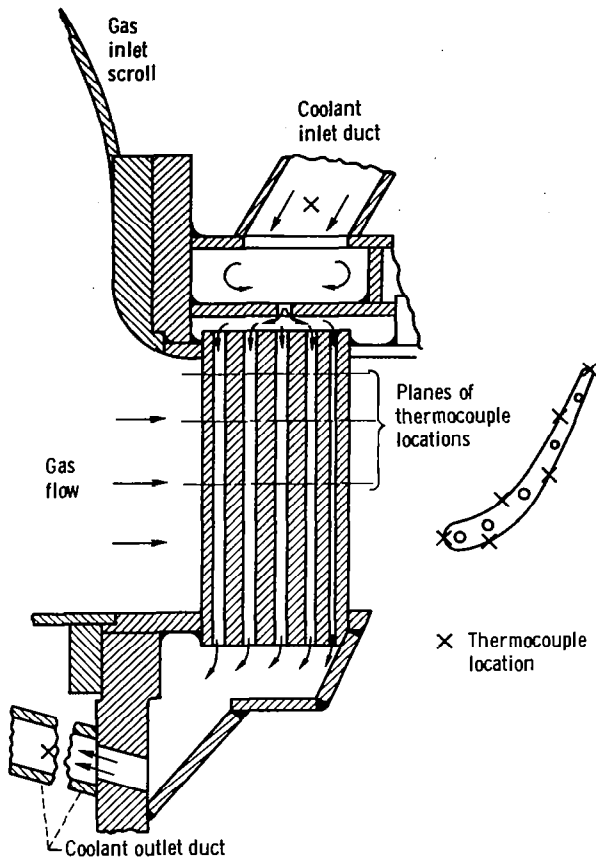


Figure 15. - NACA forced-convection water-cooled stator (ref. 4).

Stator Blade Cooling

A water-cooled stator was tested by the NACA Lewis Laboratory; figure 15 shows the cooling scheme. Results reported in Freche and Diaguila (ref. 5) indicate that high trailing edge temperatures were the only problem. Figure 15 shows that because of the hole spacing a high trailing edge temperature would be expected.

May (refs. 39 and 40) used a static cascade to test a forced-convection-cooled rotor blade. The cooling scheme consisted of four drilled holes connected by a manifold at the blade tip. Using an organic fluid as the coolant allowed operation at coolant temperatures up to 673 K (752° F) at 1.11×10^6 pascals (161 psi) without boiling in the passage. The maximum turbine inlet temperature tested was 1573 K (2372° F). The maximum measured blade temperature was 1013 K (1364° F) at the trailing edge. Thermal gradients were very high — of the order of 444 K (800° F).

Stappenbeck and Moskowitz (ref. 41) tested liquid-metal-cooled stator vanes in the rig shown schematically in figure 16. An air — liquid metal heat exchanger was included in the coolant loop to simulate recuperative heating of compressor discharge air. A maximum gas temperature of 1894 K (2950° F) was achieved with no failures; however, the total test duration was short. Current technology liquid metal pumps were found to be heavy and cumbersome.

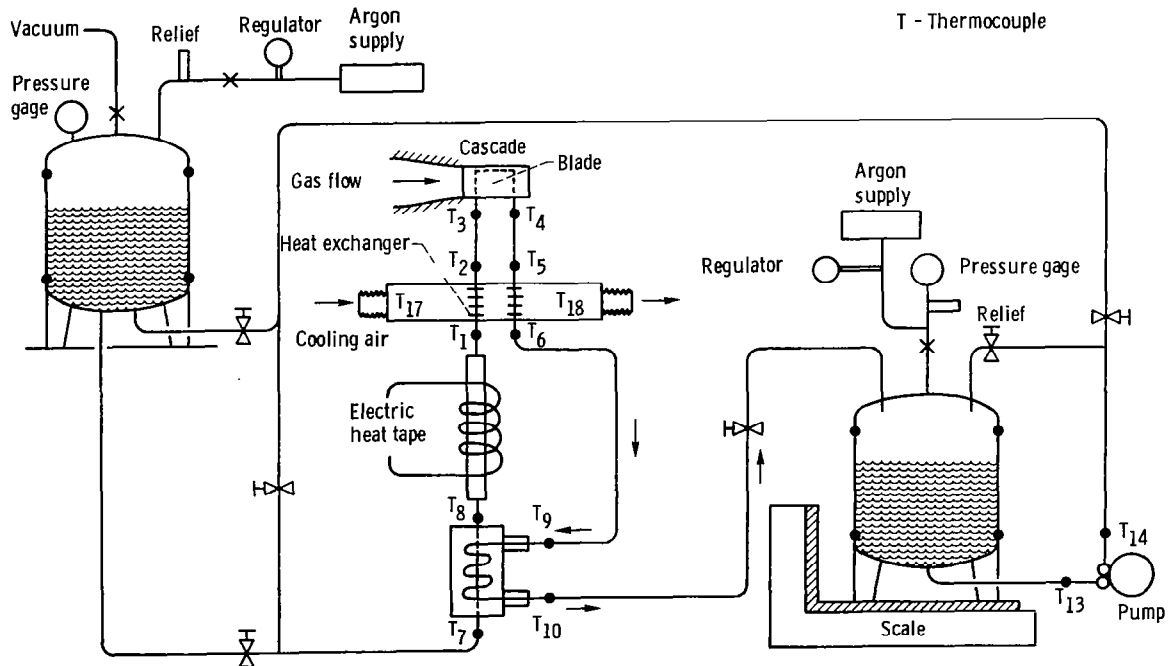


Figure 16. - Flow schematic of Curtiss-Wright liquid-metal-cooled static cascade (ref. 41).

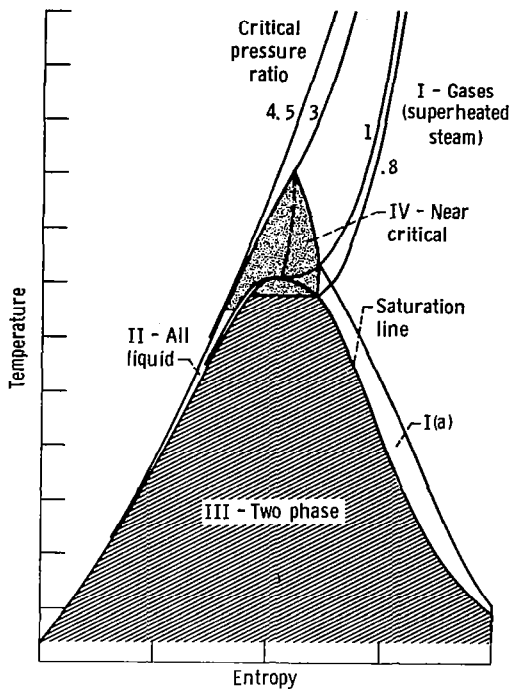


Figure 17. - Heat-transfer regions on T-S diagram.

BASIC AND SUPPORTIVE RESEARCH

In this section the physical mechanisms underlying the heat transfer and fluid flow involved in each of the cooling schemes mentioned previously are discussed. The basic and applied research that has been conducted on these mechanisms is also discussed. Forced-convection heat transfer and fluid flow processes are reviewed for each region on the temperature-entropy (T-S) diagram of the coolant. These regions were defined by Hendricks, Simoneau, and Smith (ref. 42). The discussions address the status of heat transfer, pressure drop, critical heat flux, and flow stability research and information as applied to each of the regions. Natural-convection heat transfer and fluid flow processes are reviewed for each of the three types of thermosyphons previously discussed.

Temperature-Entropy Diagram

The T-S diagram for water is shown in figure 17. Water was chosen because it may be the most practical fluid for turbine cooling applications; however, the T-S diagram of other fluids can be divided into the same regions as those to be discussed herein.

In region I the fluid is all in the gas phase; in the case of water, it is superheated steam. Region II is the all-liquid region. Region III is the two-phase

region: both liquid and gas phases are present here. Region IV is the so-called near-critical region: here large changes in thermal properties as well as some physical properties with temperature and pressure make heat transfer and fluid flow modeling difficult.

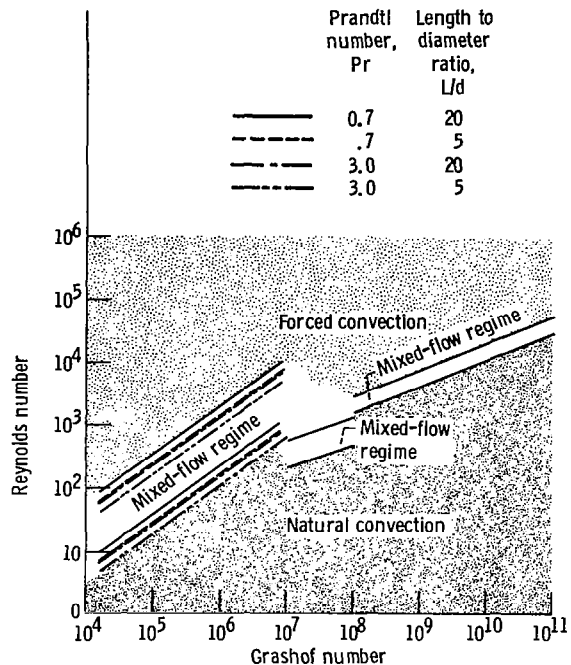


Figure 18. - Boundaries of forced, free, and mixed convection regimes (Eckert, Diaguila, and Curren, ref. 45).

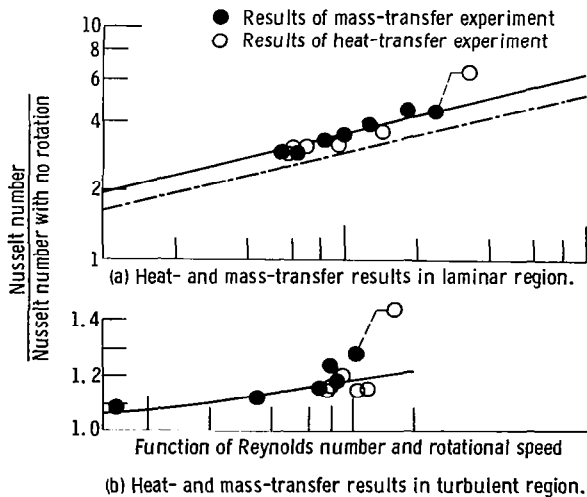


Figure 19. - Effect of rotation on laminar and turbulent heat transfer for flow in pipe (Mori et al., ref. 47).

Forced Convection

Region I (Superheated Steam)

Eckert, Diaguila, and Current (ref. 43) conducted experiments with air flowing up a vertical, 3.66-meter-(12-ft-) long stationary tube with a length to diameter ratio of 5. The test setup enabled them to obtain very large Grashof numbers. They concluded that either forced convection, natural convection, or mixed convection dominates, depending on the Reynolds and Grashof numbers as shown in figure 18. On the upper shaded portion (labeled Forced convection) of the figure, free convection contributes less than 10 percent to the total heat transfer. On the lower shaded portion (labeled Natural convection), forced convection contributes less than 10 percent to heat transfer. In the area in the middle neither mode of heat transfer is dominant. Eckert, Diaguila, and Livingood (ref. 44) recommended using the larger of either forced- or natural-convection heat-transfer coefficient predictions in the mixed regime. They also imply that data for short tubes (length to diameter ratio of about 5) can be applied to longer tubes. Even though Rayleigh numbers (product of Grashof and Prandtl numbers) were very large, these experiments did not simulate heat transfer in a rotating environment because no Coriolis forces were involved.

Mori et al. (ref. 45) conducted heat-transfer experiments on air flowing in a straight tube rotating perpendicular to its axis. Their experiments show that up to about 5 000 g's for already turbulent flow the rotation increases the heat transfer slightly (about 10 percent). For laminar flow a substantial increase in heat transfer and a nonuniform distribution of Nusselt number around the passage periphery occurs. The effect of increasing heat transfer with rotation is shown in figures 19(a) and (b) for the laminar and turbulent cases, respectively. For already turbulent flow it appears that the Dittus-Boelter correlation (ref. 46) could be used to predict heat-transfer coefficients within slightly less than 20-percent error. Klick and Wegner (ref. 47) show similar results for both air and water. They report an increase in the critical Reynolds number at which transition to turbulent flow occurs with rotation.

In summary, in region I the large acceleration present due to rotation brings free convection into play for a forced-convection configuration. Either forced convection or free convection was found to be dominant for the static case depending on the relative magnitudes of the Reynolds and Grashof numbers. Limited data for the rotating case indicate that for turbulent flow a forced-convection heat-transfer correlation could be used.

Region II (Liquid Water)

Two possible cases arise in region II. If the pressure is above the critical pressure, no local boiling will occur. If the pressure is below the critical pressure, subcooled boiling can occur.

In turbines with forced-convection cooling schemes similar to that shown in figure 1, the coolant pressure within the blades will be supercritical. If coolant flow rates are high enough, the coolant temperature will remain subcritical and all liquid flow will result. The results of Mori et al. (ref. 45) are probably true for liquids as well as gasses; thus, for turbulent flow the Dittus-Boelter correlation for heat transfer in a pipe could be used along with the results of Eckert et al. (refs. 43 and 44).

Johnson (ref. 48) used a rotating rig and simulated the cooling passages of a forced-convection water-cooled turbine. His results indicate that heat transfer at high Rayleigh numbers is less influenced by free convection than extrapolated results from lower Rayleigh numbers would predict.

Scheele, Rosen, and Hanratty (ref. 49) conducted experiments for both upflow and downflow of water in a heated tube. They observed a dye trace to determine when transition from laminar flow occurred. They found that, for both vertical upflow and downflow, laminar flow could become unstable at Reynolds numbers below the forced flow critical value if the heating rate were above a certain critical level. For upflow only, the critical heating rate was a function of the tube length to diameter ratio. This seems to be in conflict with the data of Klick and Wegner (ref. 47), but in the case of reference 49, no Coriolis forces are present. Thus, it appears that data from static rigs cannot be safely extrapolated to the rotating case to predict laminar to turbulent transition.

By Reynolds analogy, if turbulent heat transfer in rotating passages can be predicted by the pipe flow correlations, then the frictional pressure drop through the cooling passages for high-pressure liquid flow can probably be calculated from pipe flow relations. To this must be added the contribution of momentum and centrifugal pressure changes in the rotating system. The pressure loss due to friction is probably small when compared to the pressure drops due to momentum change (flow around corners and area changes) through the system.

Cooling schemes with trailing edge or tip ejection similar to those shown in figure 4 could have the coolant pressure below the critical pressure. In this case, subcooled boiling is possible unless the blade metal temperature is maintained below the incipient boiling point. The latter would require excessive coolant flow and would result in too much

heat loss from the gas turbine cycle to be practical.

Rohsenow and Hartnett (ref. 50) and Hsu and Graham (ref. 51) present good reviews of subcooled boiling literature for the nonrotating case. Rohsenow and Hartnett recommend a correlation for heat-transfer coefficients based on the sum of the calculated pool boiling and forced-convection heat fluxes. Gray et al. (ref. 52) and Marto and Gray (ref. 53) obtained boiling data for a rotating boiler at up to 475 g's. They found that boiling heat-transfer coefficients increased with acceleration at low heat fluxes but were independent of acceleration at high fluxes (fig. 20). The data in figure 20 are essentially for pool boiling at high acceleration where the g-vector was perpendicular to the heated wall; for the case of a turbine, the orientation of the g-force will be parallel to the wall. Symons and Bonamy (ref. 54) show with a theoretical model verified by photographic data that the orientation of the boiling surface with respect to flow direction has a substantial effect on cooling capacity. This is due to the change in bubble frequency. The high g-field and Coriolis forces present in the turbine case are sure to affect bubble production and dynamics and thus heat transfer. No known work has been done on subcooled boiling heat transfer in a rotating tube.

Hsu and Graham (ref. 51) show that increased

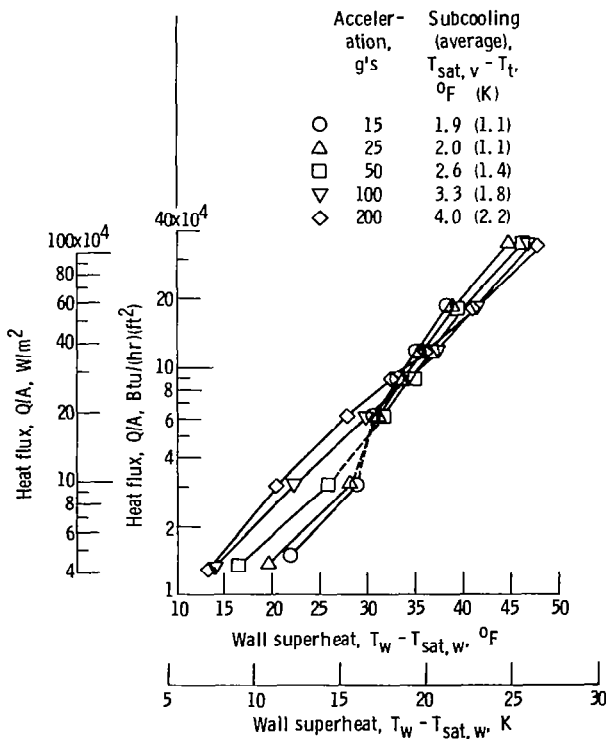


Figure 20. - Effect of gravity on heat flux as function of wall superheat (Gray and Marto, ref. 54).

acceleration has little effect on nucleate boiling heat transfer; however, their results are based on data where the heat flux and acceleration vectors are parallel and accelerations are moderate. They reason that the dominant mode of heat transfer is evaporation from a microlayer of liquid between the bubble and the heated surface and that other modes such as liquid-vapor exchange and microconvection are secondary. In the liquid-vapor exchange mechanism it is theorized that the bubbles leaving the heated surface drag hot fluid away from the wall and thus allow cool liquid to contact the wall. The microconvection theory assumes that bubble growth velocity is large enough to enhance liquid convection and produce the large heat-transfer coefficients of nucleate boiling. When the acceleration is increased to 20 000 g's and the gravity and heat flux vectors are no longer parallel, liquid-vapor exchange, microconvection, or convection alone may become dominant. Limited basic heat-transfer and passage wetting research is being conducted at the General Electric Company; some of these results are reported in reference 55.

Subcooled boiling pressure drop for low pressures is treated by Stone (ref. 56). His correlation contains an orientation parameter to account for gravity effects, but data are only available for 1 g. For high pressures Thom (ref. 57) extended the method of Martinelli and Nelson (ref. 58) for horizontal flow to include a contribution to pressure loss from gravity in vertical flow. A correlation based on vertical upflow in tubes at 1 g is presented; however, no data are available for the rotating case or for higher accelerations.

When dealing with boiling heat transfer, critical heat flux must also be considered. Critical heat flux (also termed burnout, boiling crisis, departure from nucleate boiling (DNB), onset of dry-wall boiling) occurs at high heat flux when the rate of production of vapor is so great that liquid is held away from the wall. At the onset of critical heat flux, heat-transfer coefficients can decrease by an order of magnitude. For the low quality regime Gambill (ref. 59) combines forced-convection and pool-boiling terms to obtain a correlation for critical heat flux for a wide range of conditions and geometries. Other critical heat flux correlations are reviewed by Rohsenow and Hartnett (ref. 50) and Hsu and Graham (ref. 51). Also discussed are the effects on critical heat flux of nonuniform axial heat flux, wall roughness, and swirling the flow with tube inserts. No data are available for critical flux in rotating tubes.

Another complication that arises with boiling is that of flow stability. Lowdermilk, Lanzo, and Siegel (ref. 60) performed experiments on a vertical tube

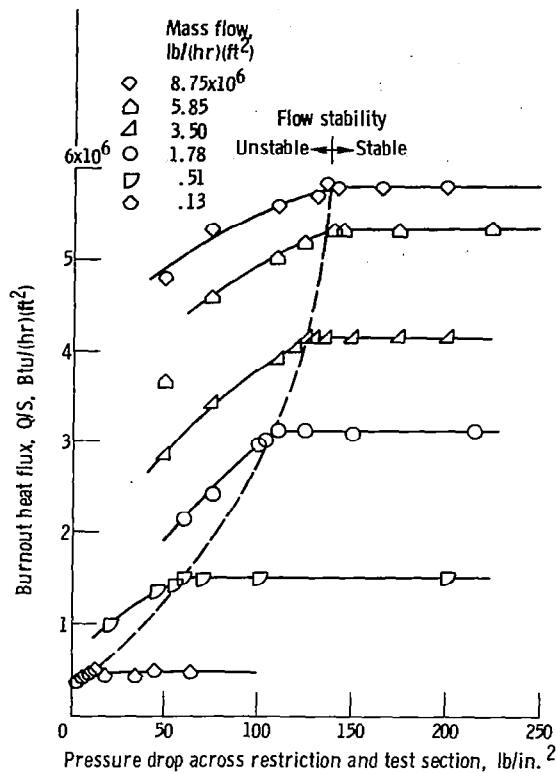


Figure 21. - Effect of flow restriction on flow stability and burnout (Lowdermilk, Lanzo, and Siegel, ref. 61).

to obtain critical heat flux data. They found that a compressible volume upstream of the heated section caused the flow to become unstable near the point of critical heat flux. Even gas bubbles in the upstream liquid could cause the instability. This instability reduced the critical heat flux considerably. They point out that flow in parallel passages would behave essentially the same as having a compressible volume upstream of the heated section; most turbine blade cooling applications would probably use parallel passages. Thus, stability may be a problem in the turbine application. They found that flow could be stabilized by restricting the inlet to the heated section, which isolated the heated section from the compressible volume. Figure 21 shows the effect of increasing pressure drop upstream of the heated section on critical heat flux. Similar results obtained by Bergles (ref. 61) show that the effects of upstream compressibility were especially destabilizing for flow in small diameter tubes. Bergles found that with stable flow small diameter tubes produced significantly higher critical heat flux than that predicted by the Gambill and several other correlations as shown in figure 22.

Gouse and Andrysiak (ref. 62) showed that instability could result only if boiling occurred in the passage. No instabilities resulted from air-water flows if bubbles were kept out of the pump. Air-water flows are often used to simulate two-phase flows. They found that decreasing the heat flux or increasing the mass flow stabilized the system.

Various two-phase flow instabilities are reviewed by Tong (ref. 63), Boure, Bergles, and Tong (ref. 64), and Hsu and Graham (ref. 51). Both static and dynamic instabilities are discussed. The most common static instability is a sudden change in flow rate to a lower value; this instability occurs, as illustrated in figure 23, when there is more than one intersection of the pump and boiling passage pres-

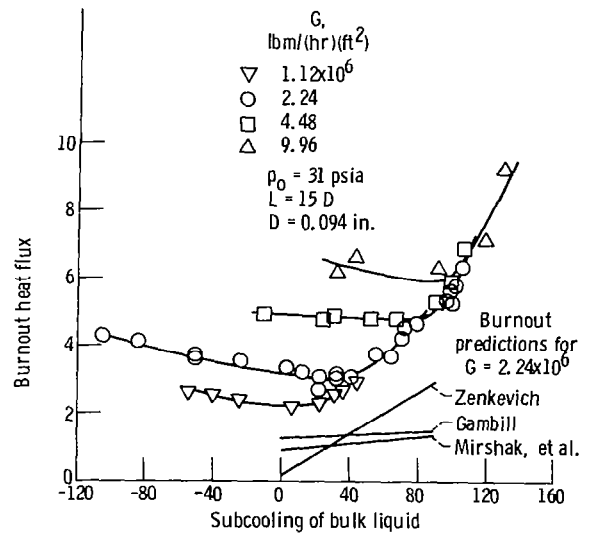


Figure 22. - Influence of velocity and subcooling in burnout heat flux (Bergles, ref. 62).

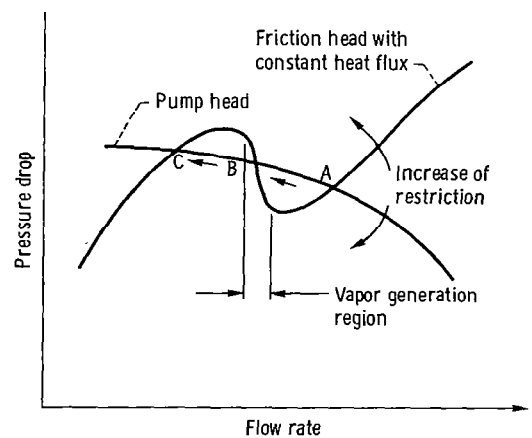


Figure 23. - Multiple intersection of flow channel and pump characteristic curves.

sure drop versus flow rate curves. The negative slope of the passage pressure drop curve is caused by vapor generation which increases the pressure drop in the passage as the flow is decreased. It is not known if this type of instability will occur in rotating passages. Associated with dynamic instabilities are acoustic waves and density waves. Acoustic waves are high frequency and generally accompany flow passage vibrations. Density waves are the most common form of dynamic instability. They can occur when a restriction downstream of the heated section passes a vapor bubble; the pressure drop is decreased because of the lower density of the vapor. Flow in the heated section then increases and less vapor is generated because the liquid spends less time in the heated section. The liquid now passes through the downstream restriction and flow decreases; this results in more vapor being generated. The cycle then repeats. The effect of rotation and high acceleration on flow stability is unknown because, among other reasons, the dynamics of bubble motion are dependent on flow configuration.

In summary, for all-liquid flow at supercritical pressure the forced-convection heat-transfer and friction correlations for turbulent pipe flow probably apply. Laminar to turbulent transition cannot be predicted from static data. For subcooled boiling, there are heat transfer, pressure drop, and critical heat flux correlations available for the static case; however, it is not known if these correlations will apply to the rotating case. Boiling flow stability could become a problem with the parallel cooling passages in a turbine blade.

Region III (Two-Phase Flow)

Operation in region III is attractive for several reasons. Because the average density of a homogeneous, two-phase mixture is less than the liquid phase alone, the pressure in the blade cooling passages will be lower than for an all liquid system. The temperature of the two-phase mixture is only a function of pressure (if equilibrium is assumed — which may not be the true case); thus, the coolant will be at nearly constant temperature as it flows through the blade. Less coolant flow is required because heat is absorbed at constant temperature by the fluid as it vaporizes. A possible drawback of using a two-phase mixture in a rotating environment is that the phases may be separated by centrifugal and Coriolis forces. Because of Coriolis forces the liquid droplets are expected to strike the walls of the feed tubes and cooling passages and be deposited there. In the cooling passages this would be desirable since an annular-type flow would benefit heat transfer, but in feed tubes it could lead to poor coolant distribution in the blade.

In a multipassage blade this could cause liquid starvation in some passages and flooding in others. It is not known if the rate of reentrainment of liquid droplets will be sufficient to compensate for the droplets that are lost by striking the walls of supply tubes.

When dealing with two-phase flows several flow patterns are possible. Heat-transfer and pressure-drop characteristics are determined by the flow pattern. Figure 24 illustrates several possible cases for flow up a vertical heated tube. At the bottom, the bulk temperature of the liquid is below the

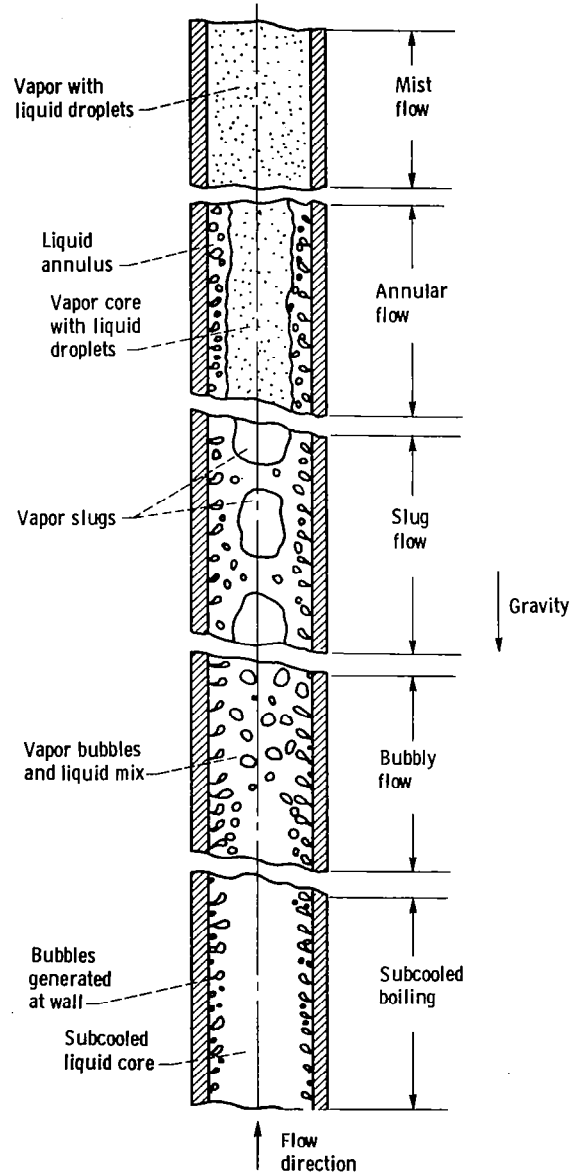


Figure 24. - Two-phase flow patterns in vertical heated channel.

saturation temperature. If the wall temperature is equal to or above the incipient boiling point, however, local boiling at the wall will take place. The vapor bubbles formed are cooled by the sub-cooled liquid and collapse. As fluid moves farther up the tube it reaches the saturation temperature. Vapor bubbles produced near the wall no longer collapse; this is the bubbly flow regime. Farther up the tube vapor bubbles agglomerate and form vapor slugs separated by liquid bridges; this is slug flow. Still further up the tube the liquid bridges disappear and annular flow results; there is an annulus of boiling water at the wall, and the core is a mixture of vapor and liquid droplets. There is a constant exchange of liquid between the annulus and the core. While droplets are striking the liquid annulus and being absorbed, new droplets are being formed from the annulus by the shearing action between the faster moving gas and the slower moving liquid. Finally, the liquid at the wall is either completely evaporated or thrown off because of vigorous boiling; this is the mist flow regime. Wall temperatures are much higher in this regime because of lower heat transfer. These flow patterns are possible for downflow and horizontal flow as well as vertical upflow; however, the conditions under which they exist will be different.

Heat transfer in the two-phase region for annular flow is treated by Chen (ref. 65) who developed a correlation for annular, vertical upflow in a tube for 1 to 70 percent quality. He combined a macroscopic term associated with forced convection and a microscopic term associated with bubble nucleation and growth. The macroscopic term is the Dittus-Boelter correlation for turbulent pipe heat transfer multiplied by a function of the Martinelli parameter (ref. 58). The microscopic term is the pool-boiling heat-transfer correlation of Forster and Zuber (ref. 66). The Chen correlation agrees well with many sets of data for water and other fluids.

Downflow boiling in a nonrotating tube was compared to that of upflow by Thorsen et al. (ref. 67). They concluded that the Martinelli method (ref. 58) could be used to calculate pressure drop in upflow but that it did not apply to downflow. They also concluded that Rohsenow's superposition method (ref. 50) of calculating forced-convection-boiling heat transfer was a valid correlation method for both upflow and downflow at low pressures but not at high pressures. For a given wall superheat, larger heat fluxes were measured for upflow than for downflow.

Stone, Gray, and Gutierrez (ref. 68) present a review of the literature and several experimental studies applied to once-through boilers. They state that in the net boiling region (the region where vapor is produced and does not recondense) cor-

relations in terms of the Martinelli parameter (ref. 58), such as Chen's correlation (ref. 65), are inadequate because they do not account for increases in the heat-transfer coefficient with pressure. They also state that nothing better exists.

There are no data available for forced-convection heat transfer in the two-phase region in a rotating environment. The large centrifugal and Coriolis forces present may cause deviation of the flow pattern from that of the stationary case and change the heat-transfer characteristics.

Martinelli and Nelson (ref. 58) proposed that two-phase pressure drop for flow in a horizontal tube could be calculated by superimposing friction and momentum change terms. Thom (ref. 57) extended the method to vertical tubes by including a gravity term. Subbotin et al. (ref. 69) made measurements of void fraction and friction for two-phase flows at high pressure that agree well with Thom's correlation. Hsu and Graham (ref. 51) give a review of methods of computing pressure drop for two-phase flows. They recommend the Thom method for high pressure, greater than 1.4×10^6 pascals (200 psia), and the Martinelli correlation for lower pressures. Thom's method can be extended to the rotating case by substituting centrifugal force for the gravity term. The validity of this extrapolation is not known since no data exist at other than 1 g.

Hsu and Graham (ref. 51) define three mechanisms for critical heat flux in two-phase flow: liquid film dryout in annular flow, dry patch under a bubble, and dryout of the film surrounding a bubble in slug flow. They reason that the different mechanisms of critical heat flux are affected by different variables. The turbine application further complicates the problem with the large centrifugal and Coriolis forces and nonuniform passage heating.

Butterworth (ref. 70) developed a critical heat flux model for circumferentially nonuniformly heated tubes. Critical heat flux was assumed to occur at the point where the liquid film on the wall dried out. Whalley, Hutchinson, and Hewitt (ref. 71) presented a similar model for the critical heat flux for vertical annular flow. The model was able to predict not only critical flux but pressure drop and void fraction. These models require a knowledge of liquid droplet deposition and entrainment rates; only limited data are available for the 1-g stationary case, and none are available for the case of a rotating tube.

Nonuniform passage heating is likely to occur in the turbine blade cooling application. Chojnowski and Wilson (ref. 72) obtained critical heat flux data for vertical upflow of water in a tube heated both uniformly and on one side. Their data indicate that critical flux may be higher for nonuniformly heated tubes. The model of Butterworth (ref. 70) predicted

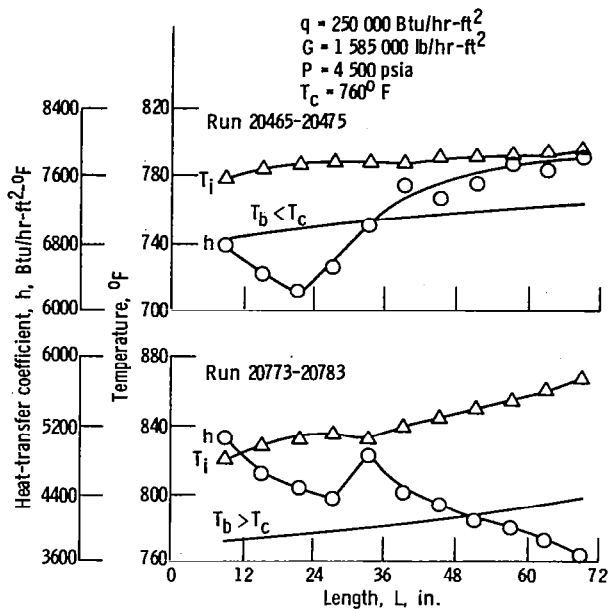


Figure 25. - "Inlet effect" for supercritical water flowing up heated tubes (Swenson et al., ref. 75).

critical fluxes that were higher than the experimental flux data.

Usenko and Fainzil'berg (ref. 73) present data for pool-boiling critical heat flux at accelerations up to 5000 g's. They found that the critical flux increased as acceleration to the 0.24 power. This trend is in agreement with the critical heat flux correlation of Gambill (ref. 59).

Many correlations based on experimental data exist; however, since these correlations do not take into account the physical mechanism of critical heat flux, they can only be used in the range of data for which they were obtained. The available correlations probably will not be valid for the passages of a rotating turbine blade. There are no critical heat flux data for a rotating passage.

As in subcooled boiling, unstable flow conditions can lead to critical heat flux at lower heat flux. Stability for two-phase flow is governed by the previously discussed criterion for subcooled boiling.

In summary, operation in the two-phase region is desirable because lower fluid pressures are possible in the blade and because coolant remains at constant temperature within the blade. Poor coolant distribution may result from the separation of the phases because of centrifugal and Coriolis forces. Heat transfer, pressure drop, and critical heat flux correlations exist for the static case; however, it is not known if these correlations can be successfully extended to the rotating case.

Region IV (Near Critical)

Schmidt (ref. 24) recommended operation in the near-critical region for turbine blade-cooling applications. His reasoning was that large property changes in this region resulted in improved natural-convection heat transfer.

Many experiments have been conducted and attempts have been made to correlate supercritical heat-transfer data. One of these investigations was done by Swenson, Carver, and Kakarala (ref. 74). They measured heat transfer for vertical upflow of water in a tube at supercritical pressures and developed a Dittus-Boelter (ref. 46) type correlation using a modified Prandtl number for "fully developed" flow. They observed large "inlet effects" which caused sharp changes in the heat-transfer coefficient. Whether the heat-transfer coefficients increased or decreased depended on whether the inlet temperature was above or below the pseudo-critical temperature. These results are shown in figure 25.

Hall (ref. 75) states that near the critical point large changes in property values can cause large buoyancy forces. He shows analytically that buoyant forces become important when the Grashoff number divided by the Reynolds number to the 2.7 power is greater than 1.16×10^5 . Hall reasons that when flow is upward (against gravity) buoyant forces at the heated wall are also upward and therefore reduce the production of turbulence with an accompanying reduction in heat transfer. With downward flow the opposite is true.

While studying the heat-transfer and coking characteristics of jet fuel at near critical conditions, Faith et al. (ref. 76) experienced large, unexpected pressure fluctuations. Pressure fluctuations of 2.4×10^6 pascals (350 psi) with a system pressure of 3.45×10^6 pascals (500 psi) were observed. Pressure fluctuations were less severe when the system pressure was 6.9×10^6 pascals (1000 psi), which is farther from the critical pressure.

Hendricks, Simoneau, and Smith (ref. 42) give a review of heat transfer near the critical region. They state that some near-critical thermal properties are not well known. They found that proximity to the critical point produces sharp changes in wall temperature; these wall temperature spikes are a function of both heat flux and mass flux. Figure 26 shows a typical curve which divides the heat flux — mass flux plane into normal and abnormal regions. In the abnormal region temperature spikes occur and heat-transfer results are not correlatable. They state that no analysis as yet is capable of modeling all aspects of near-critical behavior.

In summary, some experiments have indicated

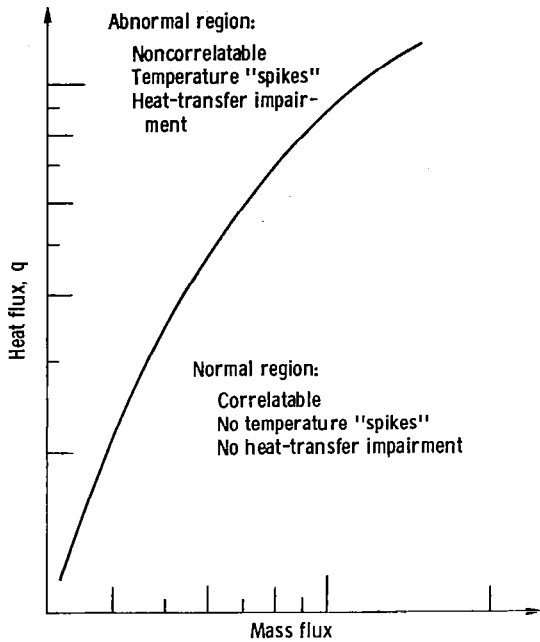


Figure 26. - Typical map of heat flux as function of mass flux (Hendricks, Simoneau, and Smith, ref. 44).

that operation in the near-critical region is desirable because of large changes in density near the critical point which would enhance heat transfer. Other work indicates that proximity to the critical point produces unexpected pressure oscillations and wall temperature spikes. No analysis is capable of modeling all aspects of near-critical behavior.

Free Convection Open Thermosyphon

Eckert and Jackson (ref. 77) used the Karman-Pohlhausen boundary-layer analysis technique to investigate the open thermosyphon analytically. They showed that thermosyphons with large length to diameter ratios did not perform as well as shorter tubes because the boundary layers of hot and cold fluid flowing in opposite directions grew together and would then mix and raise the effective coolant temperature.

Eckert and Diaguila (ref. 78) used a large, vertical heated tube in air to obtain Grashof numbers between 10^8 and 10^{13} . These large Grashof numbers simulated expected conditions in a turbine; however, the tube tested had a length to diameter ratio of about 5, which is too small for most turbine blade applications. Coriolis forces were not simulated in the tests.

Lighthill (ref. 79) used approximate solutions to the boundary-layer equations to study the flow and heat transfer in an open thermosyphon. He iden-

tified three regimes for laminar flow in the tube:

(1) For a tube with a small length to diameter ratio, the boundary layer on the tube acts the same as the boundary layer on a vertical heated plate in an infinite medium.

(2) For longer tubes, the boundary layer fills the tube and the velocity and temperature profiles are similar and only change in magnitude.

(3) For tubes with large length to diameter ratios, the flow is the same as in case (2) for a portion of the tube but near the closed end of the tube the fluid is stagnant.

Lighthill's predictions for Nusselt number as a function of Grashof number for the laminar case are shown in figure 27. These predictions agree well with the results of the experimental investigations to be discussed. He proposed the same regimes for turbulent flow, but the turbulent predictions do not agree as well with most experiments.

Martin and Cohen (ref. 80) and Martin (ref. 81) performed experiments on open thermosyphon

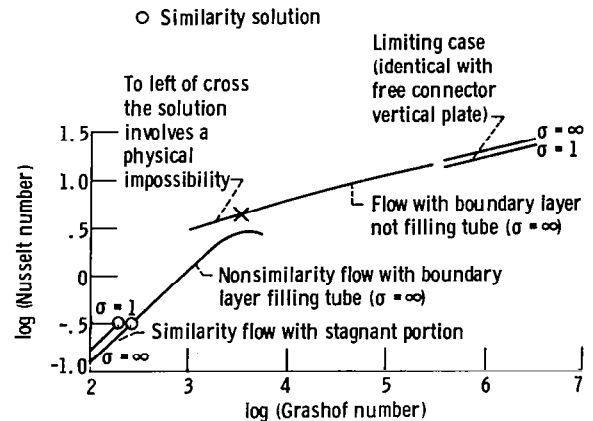


Figure 27. - Laminar predictions of Lighthill (ref. 80). (Prandtl number, σ .)

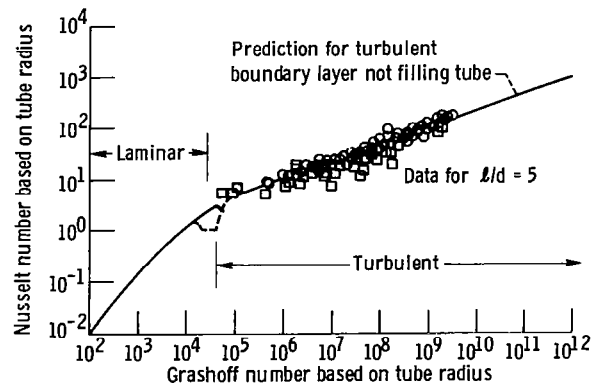


Figure 28. - Lighthill's turbulent boundary-layer-type flow prediction compared to data of Eckert and Diaguila (ref. 79).

tubes with a constant wall temperature boundary condition using several different fluids. They found that the laminar results agreed with Lighthill's theory. As predicted by Lighthill, heat transfer is impeded when the flow becomes turbulent because of the mixing of hot and cold streams. Lighthill predicted a regime of turbulent boundary-layer-type flow at very high Rayleigh numbers; however, this prediction was not confirmed experimentally. Martin claimed there was strong evidence that fully mixed flow would persist indefinitely and that this would be the predominant mode of heat transfer in the gas turbine blade.

Hartnett and Welsh (ref. 82) made measurements for a closed thermosyphon with a constant heat flux boundary condition. Average measurements agree with Lighthill's laminar theory. Average heat-transfer coefficients also agree with Martin's data for constant wall temperature. Hasegawa et al. (ref. 83) found that results obtained at constant wall temperature worked well at constant heat flux and in between the two. They found good agreement with Lighthill's laminar theory, but like other investigators they were not able to obtain the turbulent boundary-layer regime predicted by Lighthill.

For small length-to-diameter-ratio flow passages the data of Eckert and Diaguila (ref. 78) agree with the turbulent boundary-layer prediction of Lighthill. Figure 28 shows the data of Eckert and Diaguila plotted with the prediction of Lighthill. Japikse and Winter (ref. 84) suggest that improved heat transfer might be possible if a flow guide were used at the inlet to keep hot and cold fluid from mixing. Eckert and Diaguila may have been able to obtain the turbulent boundary-layer flow where other investigators did not because of their inlet conditions. Fluid did not enter Eckert and Diaguila's thermosyphon from a reservoir as in other investigations but was supplied by a separate tube. Hot fluid was taken off by other tubes around the inlet of the thermosyphon. The thermosyphon tube used by Eckert and Diaguila was relatively short when compared to those used by other investigators; this may also explain the turbulent boundary-layer-type flow.

Bayley (ref. 85) and Bayley et al. (ref. 86) concluded from an analysis that data for liquid metals taken at low Grashof numbers could be extrapolated to the turbine operating range and that turbulent results can be predicted from laminar results because of the high fluid conductivity. Bayley and Czekanski (ref. 87) obtained correlations for liquid metals in open thermosyphons with isothermal walls.

Alcock (ref. 88) suggested that the effect of Coriolis forces in the turbine might be simulated by tilting a static model; thus, a component of gravity

is used to simulate the Coriolis force. Martin and Cresswell (ref. 89) found that for fluids with Prandtl numbers about 1 heat transfer always increased with tilting angle. Heat transfer was found to vary around the perimeter of the tube with tilting. Martin (ref. 90) measured the heat transfer in an inclined thermosyphon for several fluids with Prandtl numbers ranging from about 1 to very large. He found that the heat transfer increased for the fully mixed turbulent regime for all angles of tilt. He hypothesized that tilting decreased mixing at the inlet to the tube. For laminar and transition flow, small angles of tilt decreased the heat transfer and large angles increased it. Leslie (ref. 91) found that tilting caused a decrease in the boundary-layer thickness.

Martin and Lockwood (ref. 92) studied the effect of orifice geometry and tilting by a flow visualization technique. They found that for tubes with large length to diameter ratios the fluid always enters from the floor of the reservoir. Collision of the hot and cold fluids at the inlet causes an instability. In the laminar impeded regime, the hot fluid parts to form holes through which cool fluid enters. For both laminar and turbulent boundary-layer flow the cool fluid mixes with part of the hot fluid before entering the tube. Tilting the tube stabilized the inlet flow and restored boundary-layer-type flow. Sharp edged orifices were thought to be preferable to rounded ones because they promoted a turbulent boundary layer and thus improved heat transfer.

A two-phase, boiling, open thermosyphon in a rotating rig was studied by Frea (ref. 93) up to about 100 g's. The critical heat flux was defined as the point where the vapor velocity was high enough to "hold up" the liquid flowing in the opposite direction. Critical flux increased by five over the static case at 75 g's. A simple mathematical model of annular liquid flow with gas core flow correlated data well. Kusuda and Imura (ref. 94) also modeled critical flux. Kusuda and Imura (ref. 95) conducted experiments with water in a boiling, open thermosyphon in a 1-g static test. They found that fluid at the centerline heated as it moved toward the closed end. When the saturation temperature was reached, the fluid temperature then stayed constant to the closed end. Bubbles from boiling agglomerated and then moved up the tube toward the open end in a slug "exploding" out the open end. This process caused the wall temperatures to oscillate wildly. Heat-transfer coefficients were larger than they were for pool boiling, but critical flux was considerably lower.

In summary, many experimental investigations have been conducted on the open thermosyphon in a nonrotating environment and several have been conducted with tilted tubes to simulate the Coriolis force that would be present in the rotating system.

The experimental results for vertical tubes generally agree with theory in the laminar regime; however, the predicted turbulent boundary-layer regime has been confirmed in only one experiment with a special inlet configuration. Although tilting the tube improves heat transfer, it is not known if this is a true representation of the effect of Coriolis forces. Very large heat-transfer coefficients were obtained in a rotating, boiling, open thermosyphon; however, the critical heat flux was lower than that for pool boiling.

Closed Thermosyphon

Lighthill (ref. 79) modeled the closed thermosyphon as two open thermosyphons with open ends joined: one is heated while the other is cooled. The reservoir temperature was the mixed temperature of the hot and cold streams of each.

Bayley and Lock (ref. 96) performed experiments with several fluids. They found all flow regimes in open thermosyphons can be found in closed thermosyphons as Lighthill suggested. Decreasing the ratio of heated to cooled length increased heat transfer. Bayley and Lock proposed three modes of coupling the heated and cooled thermosyphons that form the closed thermosyphon:

- (1) Conduction with high Prandtl number fluids
- (2) Convection — distinct flow streams carried hot and cold fluid from one half of the thermosyphon to the other
- (3) Turbulent mixing for low Prandtl number fluids

The authors found turbulent mixing to be the best model. They tested a device that was supposed to aid the hot and cold streams in crossing, thus avoiding the mixing of hot and cold streams, but it failed to work.

Japikse and Winter (ref. 97) found that for high Prandtl number fluids the exchange process between hot and cold streams was convective. For fluids with Prandtl numbers less than 10 turbulent mixing was the mode of transfer.

Larsen and Hartnett (ref. 98) experimented with the tilted closed thermosyphon. Their results showed that heat transfer increased with tilting. They concluded that hot fluid must flow up one side and cool fluid down the other, thus eliminating the mixing zone. Japikse, Jallouk, and Winter (ref. 99) reached similar conclusions.

Larkin (ref. 100) conducted experiments on a two-phase, boiling, closed thermosyphon using a glass tube. He noted that as the operating temperature increased the condensate boiled and was thrown off the tube wall causing dryout of the upper boiler tube segment. Tilting the tube showed no adverse effects. If liquid was carried by the vapor

to the condenser it hindered condensation.

Wallis and Makkenchery (ref. 101) tested tubes with downflow of a liquid annulus and upflow of a gas core to determine the critical velocity. This is the velocity at which the gas core "holds up" the liquid annulus. This simulated the boiler end of a two-phase closed thermosyphon. If the flow is "held up," then the flow of liquid coolant to the blade tip would be stopped. They found that when the dimensionless diameter (nondimensionalized by multiplying by the square root of gravity times density difference between the phases and divided by surface tension) is less than 2 the critical velocity tends to zero. These results indicate that for turbine applications where the dimensionless diameter is likely to be between 20 and 50 reasonable velocities are attainable without reaching the critical heat flux.

After a review of all liquid-cooling systems, Cohen and Bayley (ref. 15) concluded that the two-phase closed thermosyphon was the most practical system for turbine cooling application. They conducted experiments on a rotating rig as mentioned earlier. Bayley and Bell (ref. 16) made heat-transfer measurements on a rotating two-phase thermosyphon filled 0, 1/4, 1/2, 3/4, and full of mercury up to 440 g's. Results are shown in figure 29, which is from reference 15. After a certain filling quantity is reached, the maximum heat flow capacity is independent of the amount of mercury in the tube. Variations in heat transfer for different rotational speeds and filling quantities could be partially explained by the variation in saturation pressure along the tube.

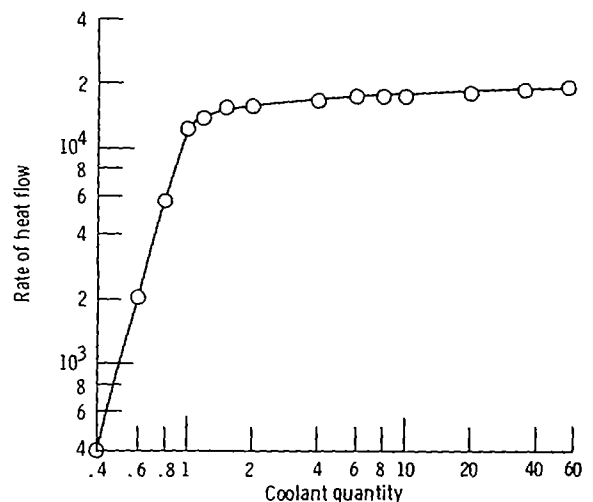


Figure 29. - Heat flow plotted against quantity of coolant for rotating, closed, two-phase thermosyphon (Cohen and Bayley, ref. 15).

Chato (ref. 102) solved momentum and energy equations on a flat plate for condensing liquid in a variable g-field. This simulated the condenser end of a two-phase closed thermosyphon. Results show that film thickness and Nusselt number approach limiting values as radial distance increases. This indicated a limit to the length of the two-phase closed thermosyphon.

In summary, the closed thermosyphon has been modeled as two open thermosyphons connected at the open ends. The method of exchange of hot and cold fluids at the junction has been shown to be turbulent mixing for most practical applications. Tilting the closed thermosyphon improves heat transfer. The two-phase closed thermosyphon has been tested in a rotating environment, and the heat-transfer capacity was found to be independent of the filling quantity.

Closed-Loop Thermosyphon

Chato (ref. 103) and Chato and Lawrence (ref. 104) used one-dimensional mass, momentum, and energy equations to develop a mathematical model to solve for the flow distribution in n-parallel channels. The channels were allowed to have different but uniform heat rates; thus, some channels could be cooled while others were heated. This would cause a density difference between heated and cooled channels which would lead to circulation. He assumed that a forced-convection friction factor could be applied since the flow in each channel would be in one direction (up some channels and down others). Predictions for a three-channel system in which one channel is unheated and the other two are heated at different rates showed that several flow patterns were possible. In the first flow pattern, flow was up in the channel with the largest heating rate and down in the other two. For the second case, flow was up in both heated channels, but flow in the channel with the lower heating rate was at a very low rate. Experiments were performed to confirm this analysis. Chato found that steady oscillation could develop in the system because of periodic boiling and geysering in the heated channels. This indicates that great care must be taken in designing cooled turbine blades with multiple passages.

Davies and Morris (ref. 105) coined the term "mixed convection thermosyphon" to describe the closed-loop thermosyphon. Mixed convection in this case meant the flow was driven by natural-convection-type density gradients but was otherwise treated as a forced flow. They conducted experiments on a rotating rig where heat was added to a leg rotating parallel to the axis of rotation and extracted in a leg that was coincident with the axis.

They concluded that transition to turbulence occurred much sooner than it did for forced convection. They developed an empirical correlation for Nusselt number based on an analysis similar to Chato's. Data were taken only from 0 to 15 g's. Japikse and Hoschendler (ref. 106) used a model similar to Chato's to design a mixed convection, cooled turbine blade.

Hayama (refs. 107 and 108) studied natural circulation in a boiling loop with heat added in the up leg. He concluded that the driving potential was due to voids in the boiling leg which decreased the average density.

Keller (ref. 109) used time-dependent equations for an inertialess system consisting of two parallel vertical tubes joined by horizontal tubes at the top and bottom. Heat was added and removed at points on the bottom and top, respectively. He found that a periodic solution exists; this indicates that the flow can oscillate. Welander (ref. 110) used linear stability theory to develop a neutral stability curve in terms of dimensionless gravity and friction coefficients. His results indicate that no unstable systems with exponential growth exist and that oscillation depends on friction and density gradients and not on system mass. Creveling et al. (ref. 111) performed experiments on a torus of water with heat added at the bottom and taken off at the top. Oscillations predicted by linear stability theory were found to exist. Hayama (refs. 107 and 108) used a stability analysis on the boiling natural-convection loop and concluded that instability could exist if the slope of the pressure-drop flow rate curve was negative near the operating point. The addition of a pump or orifice to the circuit stabilized the flow. The implication of flow oscillation to heat transfer in the turbine application is not known.

In summary, the closed-loop thermosyphon has been modeled by assuming that forced-flow heat transfer and pressure loss correlations apply to free-convection-driven flows in a closed loop. This analysis has not been confirmed by experiment. It has been shown by analysis and confirmed by experiment that flow oscillation in the closed-loop thermosyphon is possible in a static system. The consequences of oscillation to heat transfer in the turbine application are not known.

TECHNOLOGY REQUIRED

This review of liquid-cooling literature has pointed out the lack of basic data and understanding of heat transfer, pressure drop, critical heat flux, and flow stability in the cooling passages of a rotating turbine blade. In this section, areas of

research required to advance our understanding of these processes are outlined.

Additional experimental heat-transfer and flow data must be obtained and correlations and prediction methods must be developed for single- and two-phase flow particularly under high centrifugal accelerations. Some two-phase heat-transfer data on stationary tubes, limited pool boiling data to about 475 g's, and single-phase flows in a rotating tube up to 5000 g's are available; however, it is not known if these data can be successfully applied to rotating systems where centrifugal accelerations may reach 20 000 g's and where large Coriolis forces are also present. There are no data for flow toward the axis of rotation, only for radial outflow. Information is also needed on the effect of non-uniform passage heating and high centrifugal accelerations on critical heat flux. The information that is available is sparse and only for nonrotating conditions. Pool boiling critical flux data are available for accelerations to 5000 g's. No forced-convection boiling critical flux data are available at high accelerations or with Coriolis forces. The effect of large accelerations on flow stability in boiling channels is also unknown. Research must also be conducted to determine the distribution of coolant phases in liquid-vapor systems. Centrifugal and Coriolis forces may cause some of the cooling passages to become liquid starved, thus causing local overheating of the turbine blade. Information must also be obtained on the effect of variables such as coolant viscosity, surface tension, and passage diameter on the wetting of the passage periphery under the influence of Coriolis forces and their combined effect on flow and heat transfer.

Heat transfer to supercritical water must be investigated under the influence of high-g forces. Correlations are available for heat transfer to supercritical water in stationary tubes, but it is not known if these correlations will be valid in a high-g environment. Test data for liquid flow in a single, heated, nonrotating tube at supercritical pressures and temperatures showed large pressure fluctuations at some conditions of operation which were not anticipated prior to testing. The problem of flow stability of supercritical fluids in rotating passages must be explored experimentally for turbine operating conditions.

The closed-loop thermosyphon and the open thermosyphon with passages connected together at the blade tip seem to be the most practical among the various thermosyphons for turbine applications because they permit using the large passage length to diameter ratios required to minimize stresses. The mixed convection analysis, where heat transfer and friction pressure drops are evaluated from forced-convection correlations for

a free-convection-driven flow, needs to be verified by experiment. The possibility of flow oscillation and the consequences to heat transfer must be investigated. The problems of flow stability of near-critical fluids in rotating passages must be explored experimentally for expected turbine operating conditions. There is no known analysis capable of modeling all aspects of supercritical heat transfer in the static case let alone the rotating case.

Other areas related to liquid cooling where technology needs to be broadened include experimental confirmation of the expected influence of highly cooled walls on combustion-gas-stream boundary-layer development and heat transfer. A currently available two-dimensional boundary-layer program predicts early transition to turbulent flow and thus higher heat transfer for highly cooled walls. Since this effect could cause higher metal temperatures than expected if not accounted for in a design, it should be investigated experimentally. The clogging of small cooling passages by deposits from cooling water needs to be evaluated as well as the methods of minimizing erosion of cooling passages and water recovery systems. Long life seals need to be developed to get coolant on and off rotating turbine wheels. Also, improved manufacturing and nondestructive inspection techniques need to be developed that allow the economical production of blades, vanes, and associated hardware that will be capable of containing coolant at the high pressures required for the long lifetimes of large industrial gas turbines.

CONCLUDING REMARKS

Although a number of liquid-cooled turbine demonstrators have been tested and some basic and supportive liquid-cooling research has been conducted, existing technology is not adequate to design a liquid-cooled turbine with a high assurance of success or to optimize liquid-cooling systems for a given turbine application. Considerable basic and applied research on the liquid cooling of gas turbines is still required.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 28, 1978,
778-11.

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1. Report No. NASA RP-1038 AVRADCOM TR 78-21	2. Government Accession No.	3. Recipient's Catalog No.	
4. Title and Subtitle REVIEW AND STATUS OF LIQUID-COOLING TECHNOLOGY FOR GAS TURBINES		5. Report Date April 1979	6. Performing Organization Code
		8. Performing Organization Report No. E-9517	10. Work Unit No.
7. Author(s) G. James Van Fossen, Jr., and Francis S. Stepka		11. Contract or Grant No.	13. Type of Report and Period Covered Reference Publication
9. Performing Organization Name and Address NASA Lewis Research Center and AVRADCOM Research and Technology Laboratories Cleveland, Ohio 44135		14. Sponsoring Agency Code	
		12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546 and U.S. Army Aviation Research and Development Command, St. Louis, Mo. 63166	
15. Supplementary Notes G. James Van Fossen, Jr., AVRADCOM Research and Technology Laboratories; Francis S. Stepka, Lewis Research Center.			
16. Abstract A review was conducted of liquid-cooled turbine technology. Selected liquid-cooled systems and methods are presented along with an assessment of the current technology status and requirements. A comprehensive bibliography is presented.			
17. Key Words (Suggested by Author(s)) Liquid cooling Gas turbines Heat transfer Turbine blades		18. Distribution Statement Unclassified - unlimited STAR Category 34	
19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 29	22. Price* A03

* For sale by the National Technical Information Service, Springfield, Virginia 22161

NASA-Langley, 1979